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SOME ASPECTS OF THE DESIGN OF LARGE HIGH-PRESSURE STEEL VALVES.

BY FRANK D. COTTERMAN * AND R. E. FALLS.*

The design of a valve derives first from its function as a regulator of fluid flow. The resulting shape, viewed as a pressure vessel with superimposed mechanical and thermal loading, defies accurate stress calculations. In Part I, the authors describe a successful method of supplementing approximate calculations by tests in order to determine the metal thickness for safe pressure-temperature rating.

Within limits, the shape of a valve may be altered to reduce pressure drop without interfering with its effectiveness as a fluid flow regulator. In Part II, the authors present test results which compare the resistance to steam flow in valves of several types.

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Part I—Determination of Metal Thickness and Rating.

The accurate determination of the proper metal thickness and the working stresses in valve bodies and bonnets is, in general, an exceedingly difficult problem. Valves vary in shape from the relatively simple contours of the conventional gate type to the more complicated form of the globe valve with the stem set at an acute angle with the run. Loading conditions may be extremely complex; for a valve body may be subjected not only to stresses induced by the internal pressure and flange bolting, but also to loads produced by improperly supported connecting piping. These loads must be resisted by the valve without distortion sufficient to cause leakage and without inducing stresses high enough to cause ultimate failure. The problem becomes particularly complex when valves for high-temperature service are considered, because of the presence of creep and the changes which take place in the properties of the materials.

Considerable work of both a theoretical and an experimental nature has been performed on flanged joints, and has resulted in the formulation of codes, such as the American Standards Association B16e (1939) Standard for Steel Pipe Flanges and Flanged Fittings. These standards specify in detail the pressure-temperature ratings, minimum body and flange metal thicknesses, number and size of bolts, and other dimensions for steel flanges and flanged fittings. However, no information has been published by means of which the stresses throughout a complex shaped valve body can be accurately calculated.

Existing formulæ:

For the design of cylinders subjected to internal pressure the formulæ of Barlow, Lame, Clavarino, and Birnie are well known, but are theoretically correct only for cylinders. Where bends or outlets, such as exist in valve bodies or elbow shapes, are encountered, modifications of these formulæ may be used. Modified Barlow's formulæ, with 50 per cent added to the thicknesses determined from them "to compensate for the shape of the fittings," have been used as the basis for the minimum body metal thicknesses specified for fittings in ASA B16e. These formulæ are as follows:

For pipes having nominal diameters of from ¼ to 5 inches,

$$P = \frac{2S}{D} (t - 0.065) - 125.$$

For pipes of nominal diameters over 5 inches,

$$P = \frac{2S}{D} (t - 0.1),$$

in which

P = working pressure, pounds per square inch.
t = thickness of wall of pipe, inches.
D = actual outside diameter of pipe, inches.
S = 7000 pounds per square inch.

The above formulæ are based on the use of carbon or carbon-molybdenum steels and a working pressure equal to the nominal or primary service pressure at the temperature rating specified in the standards. Pressure ratings for other temperatures have also been established by increasing or decreasing the working pressures in proportion to the working stresses permitted at the various temperatures.

With the substitution of the proper shape compensation factor and the proper working stress for other materials, these modified Barlow's formulæ may be employed to determine the metal thickness in a valve body. Variations in the calculated thickness must, of course, be made to provide ample fillets, to allow for possible corrosion or oxidation at high temperatures, and to produce sound castings as dictated by foundry requirements.

WORKING STRESSES:

Safe working stresses for a given material depend upon its physical characteristics at the operating temperature, which is only slightly less than that of the fluid in a well-insulated valve. Recently, Table P-9 in the ASME Power Boiler Code has been revised to include the maximum allowable working stresses for various ferrous materials for metal temperatures from —10 degrees F. to 1000 degrees F. Figure 1 portrays in graphical form the values given for the carbon and carbon-molybdenum steels

commonly used for valves, flanges, and flanged fittings. Below temperatures of 650 degrees F. for carbon steel and 800 degrees F. for carbon-molybdenum steel, the allowable working stresses shown in Figure 1 are equivalent to 1/5 of the nominal tensile strength for forged materials and 1/7 for cast materials. Above

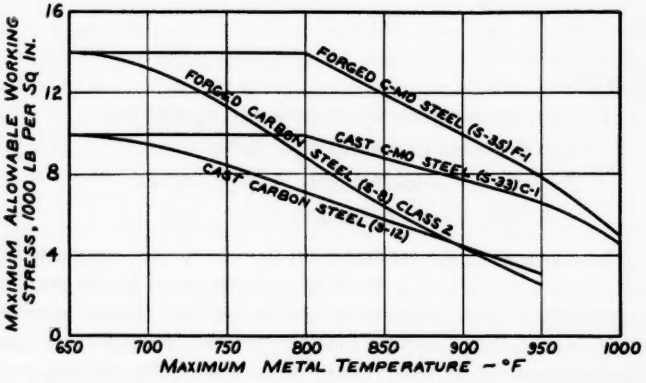


FIGURE 1.—ASME POWER BOILER CODE SPECIFICATIONS (TABLE P-9, REVISED AUGUST 18, 1939) FOR ALLOWABLE WORKING STRESSES.

these temperatures, the values are based on available creep data and general experience. Values for high-temperature service for other alloys, which are not included in this table, are based upon similar considerations of their individual characteristics.

RATING TESTS:

In order to determine whether or not a design is safe, the internal hydrostatic pressure at which the proportional limit of the valve material is reached may be determined by a test at room temperature, in which a series of simultaneous observations of internal hydrostatic pressure and deflection are made. This method is applicable only to materials which have a definite proportional limit. Valves made of cast iron, for example, are generally subjected to bursting pressure tests to determine their ultimate strengths. Tests to destruction, however, are not generally applicable to steel valves, because of the difficulty experienced in

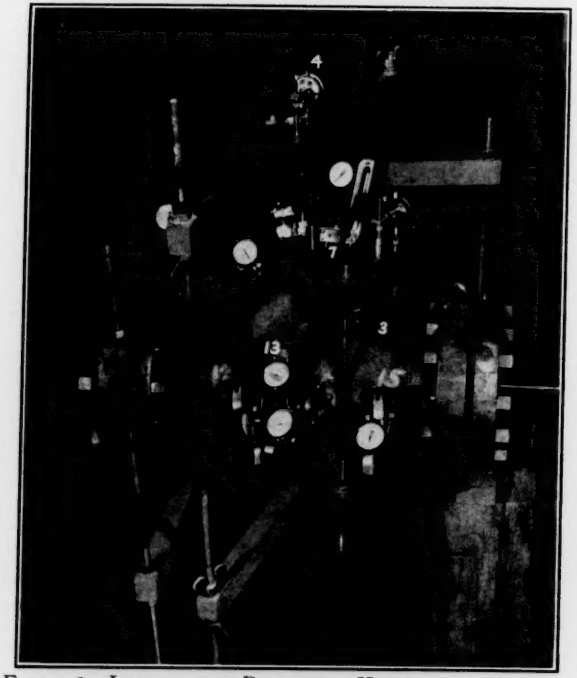


FIGURE 2.—LOCATION OF DIALS FOR HYDROSTATIC PRESSURE TEST ON 6-INCH 600-POUND STEEL GLOBE VALVE OF IMPROVED DESIGN.

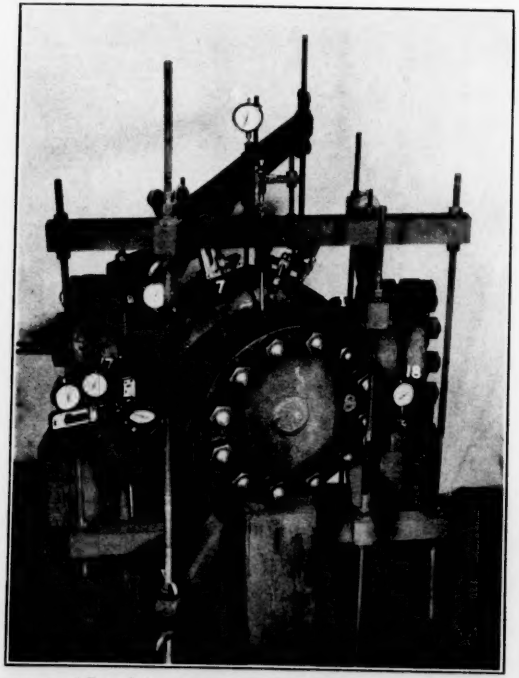


FIGURE 2 (CONT'D).—LOCATION OF DIALS FOR HYDROSTATIC PRESSURE TEST ON 6-INCH 600-POUND STEEL GLOBE VALVE OF IMPROVED DESIGN.

holding flanged joints tight at high pressure, and the fact that, in a ductile material, yielding of the material under high stresses may mask the true location of the maximum stress.

The procedure for determining the proportional limit under internal hydrostatic pressure may be illustrated by tests conducted on a 6-inch 600-pound Navy globe valve of improved design which will be described in Part II of this article. The end ports of the valve were blanked off with blind flanges, one of which was tapped to admit pressure. Deflectometers and strain gages of 2-inch span, equipped with 0.001-inch or 0.0001-inch dial indicators, were mounted at 18 strategic positions on the valve body, as shown in Figure 2. The location of these points was determined primarily on the basis of experience, since lime, Portland cement, plaster of Paris, and similar "brittle" coatings have not proven very satisfactory for locating the maximum stress in such structures, because of the large deflections necessary to produce flaking or cracking of the coating.

In conducting the test, the pressure was increased to a maximum of 4600 pounds per square inch, and deflections were read and plotted at each 200 pounds per square inch increment. The test was discontinued at this pressure since it was obvious that the proportional limit had been exceeded.

The pressure was then released and the permanent set shown by some of the dials was determined. This method has been found to give more reliable results, for the small magnitude of deflections encountered, than that of reducing the pressure to zero and recording the permanent set after each increment of pressure has been applied.

Figure 3 shows the data obtained from several representative dials. The lowest proportional limit, as determined by the point at which the linear relationship between pressure and deflection ceased to exist, evidently occurred at 3100 pounds per square inch hydrostatic pressure in the crotch of one of the elbow-shaped portions of the valve (Dial 18).

To obtain the safe working pressure, Paragraph A-30 (revised August 18, 1939) of the ASME Power Boiler Code recommends the use of the formula:

$$P = HS/E$$

in which

P = maximum allowable working pressure, pounds per square inch.

H = hydrostatic pressure at the proportional limit of the pressure part, pounds per square inch.

S = safe working stress permitted for the material at the maximum operating temperature.

E = average proportional limit of the material, pounds per square inch.

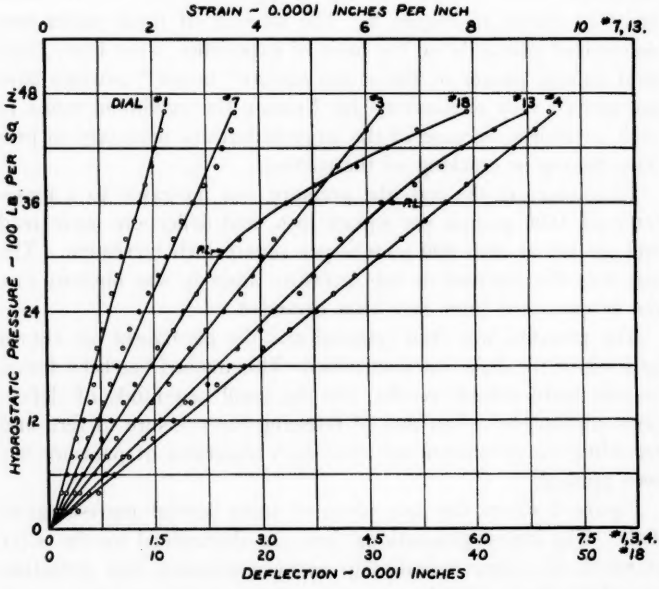


FIGURE 3.—PRESSURE-DEFLECTION CURVES FROM HYDROSTATIC PRESSURE TEST ON 6-INCH 600-POUND STEEL GLOBE VALVE OF IMPROVED DESIGN. (FIGURES AT END OF EACH ABSCISSA SCALE REFER TO DIAL NUMBERS.)

The logic upon which this formula is based will be apparent if it is rewritten in the form $\frac{P}{H} = \frac{S}{E}$. The left side of this equa-

tion then represents the ratio of the allowable working pressure to the pressure at the proportional limit for the valve, while the right side represents the ratio of allowable working stress to the stress at the proportional limit for the material from which the valve is constructed.

The average proportional limit of the material may be obtained from tensile tests conducted on a number of coupons machined from the valve after completing the hydrostatic pressure test, provided that such bars are cut from a section in which the proportional limit has not already been exceeded. Results obtained from five bars cut from various sections of the body of the valve tested gave an average proportional limit of 46,000 pounds per square inch for the cast carbon-molybdenum steel (Grade C-1) from which the body was made.

Since the valve tested was designed for 850 degrees F. steam service, it can be assumed, by reference to Figure 1, that the maximum allowable working stress is 8800 pounds per square inch, which corresponds to a metal temperature of 850 degrees F.

Substitution of this value, along with those previously determined for H and E, into the ASME Code formula, gives

$$P = \frac{HS}{E} = \frac{3100 \times 8800}{46{,}000} = 593 \text{ pounds per square inch}$$

safe working pressure. When it is considered that the actual operating temperature of the metal will probably be somewhat less than the 850 degrees F. rated steam temperature, and that the rate of loading the tensile bars to find the proportional limit of the material was considerably greater than that used during the hydrostatic pressure test of the valve body, it is evident that the calculated 593 pounds per square inch safe working pressure may be increased slightly. The valve may therefore be rated safely for 600 pounds per square inch working pressure at 850 degrees F.

CONCLUSION:

In this discussion it has been shown that the method of determining the metal thicknesses and the safe pressure-temperature rating for steel valves is not an exact process. However, the procedure described has been found to be satisfactory over a period

of years and it will probably continue to be used until sufficient research is performed to give a rational or more exact method of design.

Part II—Design in Relation to Pressure Drop.

The pressure drop caused by valves and fittings in steam lines is a matter of concern to piping engineers because it is always accompanied by an increase in the entropy of the fluid and by a decrease in the energy available for doing work. Consequently, the piping engineer attempts to keep the pressure drop between the boiler and the turbine throttle as low as possible but the cost of the piping, and often its weight, are limiting factors.

In main steam lines it is customary to use only gate valves—with the exception of the automatic non-return valve—because the resistance this type of valve offers to flow is very small. Although globe valves offer appreciable resistance to flow, there are some installations in which they must be used. In such cases it is essential that the designer have data available for accurately estimating the pressure drop which will occur. Therefore, in order to obtain more complete information on the characteristics of flow through globe and angle valves and also to effect improvements on certain U. S. Navy design valves, the Crane Co. Research and Development Laboratories conducted a series of flow tests on 6-inch 600-pound cast steel valves.

VALVES TESTED:

In Figure 4 are shown two views of a 6-inch 600-pound globe valve design which was in use prior to these tests. In this valve four integrally cast reinforcing struts connect the bridge wall with the body neck. The valve disc is guided by a detachable liner.

As a result of the tests the design shown in Figure 5 was developed. In this improved design there are only three reinforcing struts and they are spaced so as to offer a minimum of resistance to flow. Since no liner is used, the disc is guided by the smooth bore of the body neck.

In internal construction, this angle valve design is similar to the globe valve design. Consequently, the same changes were

made in the angle valves as were made in the globe valves; that is, the number and spacing of the reinforcing struts was altered and, in the improved design, the liner was removed.

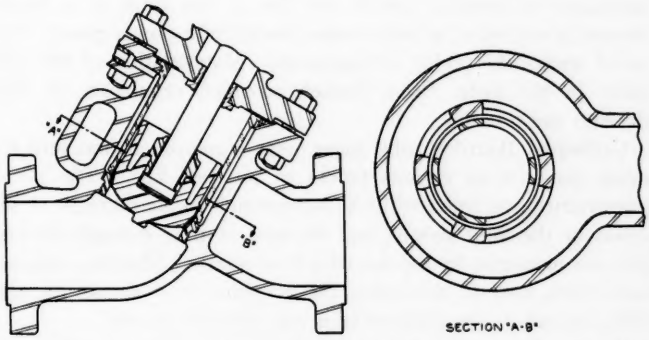


FIGURE 4.—INTERNAL CONSTRUCTION OF 6-INCH 600-POUND GLOBE VALVE—ORIGINAL DESIGN.

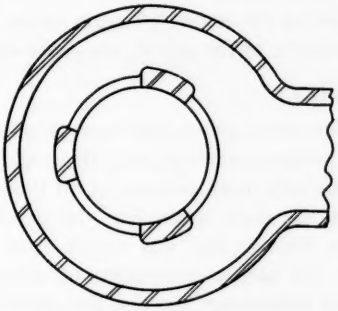


FIGURE 5.—IMPROVED DESIGN GLOBE VALVE—SECTIONAL VIEW SAME AS SECTION "A-B" IN FIGURE 4.

In addition to the design valves described, flow tests of a conventional gate valve and a horizontal swing check valve were made at the same time.

TESTING APPARATUS:

The adequate testing of these 6-inch valves with 600 pounds per square inch steam demanded a flow of at least 100,000 pounds of steam per hour. In order to obtain this rate of flow it was necessary to install a special test line in the form of a bypass around a section of a main steam header in a power plant. Figure 6 shows the piping arrangement. Manipulation of the gate valve in the main steam header provided regulation of flow through the test line.

Calibrated Bourdon tube gages were employed to measure the steam pressure at various points in the test line. The steam temperature was indicated by a thermocouple located directly in the center of the fluid stream, and the rate of flow through the test line was measured by the use of a flow nozzle. Mercury manometers were used to determine the pressure drop across the test valve, as well as the differential across the flow nozzle.

During these tests the average steam conditions at the flow nozzle were 615 pounds per square inch pressure and 730 degrees F. total temperature. Although the test line was well insulated, the test valves were entirely un-insulated. The piping shown in Figure 6 was used in the testing of angle valves; for globe valve testing a slight change in the piping was necessary.

TESTING PROCEDURE:

All of the globe and angle valves were tested in two positions, that is, with the pressure admitted both above and below the discs, which were in the wide open position at all times. By operating the gate valve in the main steam line, the mean velocity of the steam flowing in the test line was varied from 1000 to 12,000 feet per minute. At various increments of velocity, as indicated by the flow nozzle manometer, the test line pressure and temperature, the flow nozzle differential, and the pressure drop across the test valve were measured.

RESULTS:

The results of these tests are shown in Figures 7 and 8, in which the pressure drop across the test valves is plotted as a function of the mean velocity of the steam in the test line. The

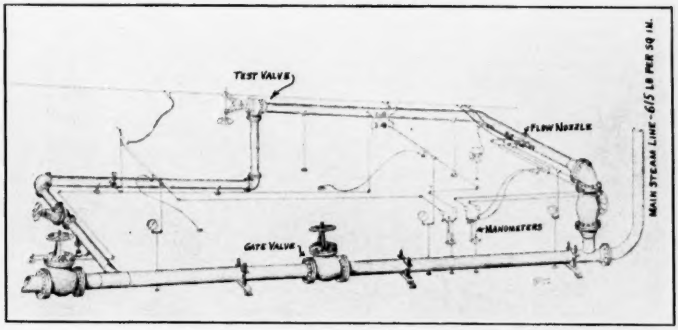
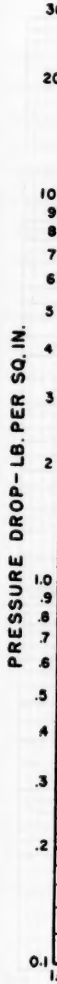


FIGURE 6.—PIPING ARRANGEMENT FOR FLOW TESTS.



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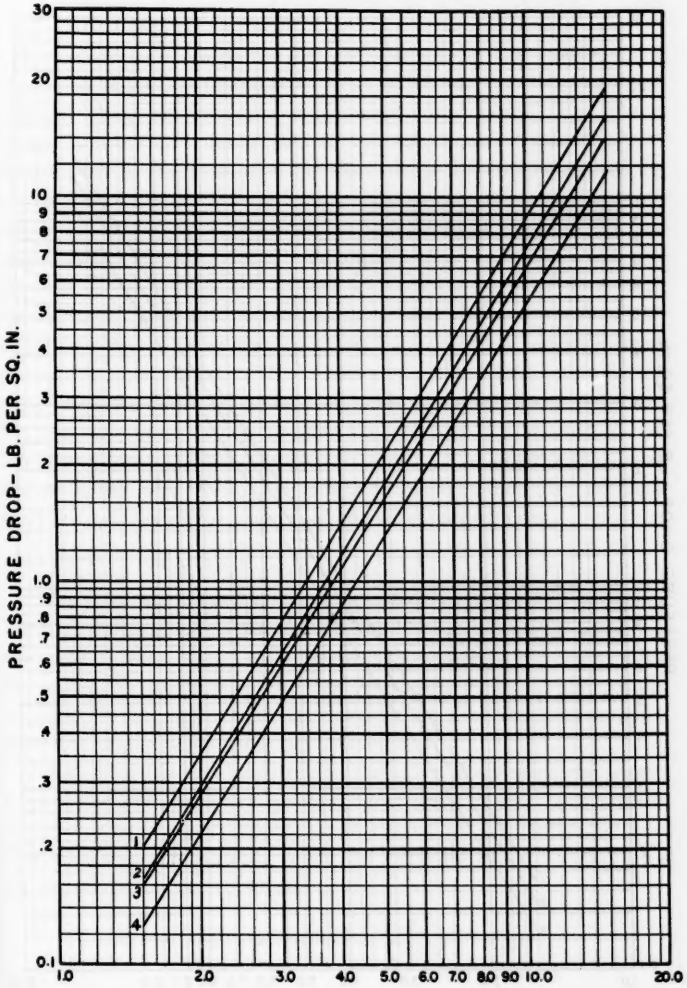

FIGURE 7.—PRESSURE DROP IN 6-INCH 600-POUND CAST STEEL GLOBE VALVES. AVERAGE STEAM CONDITIONS: 615 POUNDS PER SQUARE INCH, 730 F.

Legend—

1. Navy Globe Valve—Original Design. Pressure admitted under disc.
2. Navy Globe Valve—Original Design. Pressure admitted above disc.
3. Navy Globe Valve—Improved Design. Pressure admitted under disc.
4. Navy Globe Valve—Improved Design. Pressure admitted above disc.

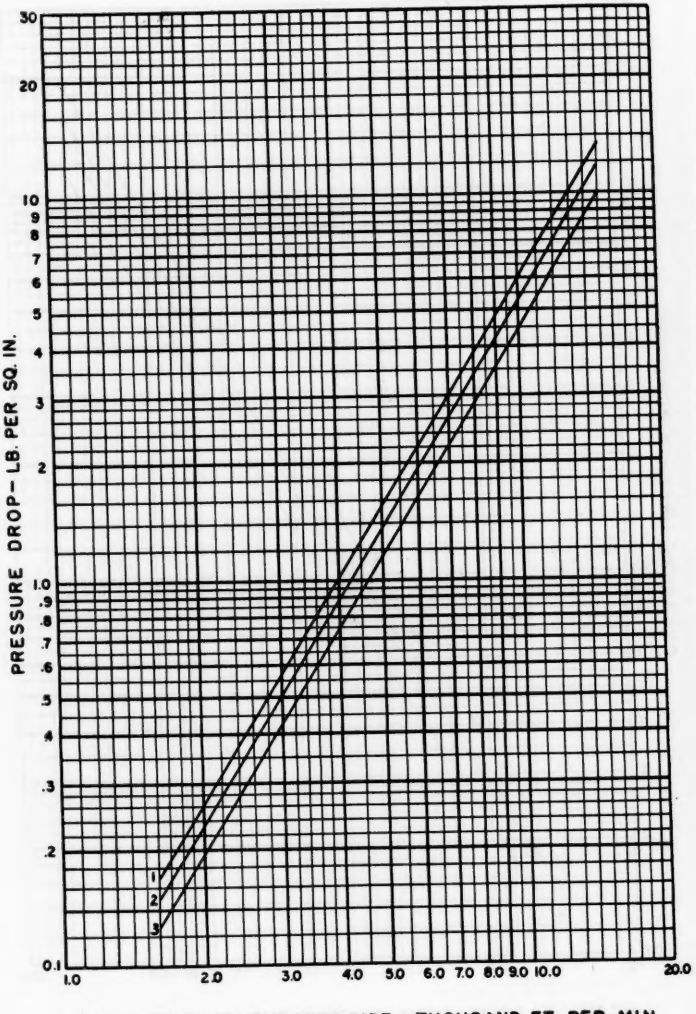

FIGURE 8.—PRESSURE DROP IN 6-INCH 600-POUND CAST STEEL ANGLE VALVES. AVERAGE STEAM CONDITIONS: 615 POUNDS PER SQUARE INCH, 730 F.

Legend—

1. Navy Angle Valve—Original Design. Pressure admitted either above or under disc.
2. Navy Angle Valve—Improved Design. Pressure admitted above disc.
3. Navy Angle Valve—Improved Design. Pressure admitted under disc.

values plotted represent the net loss caused by the valves only: the losses in the test piping have been deducted from the measured pressure drop between the manometer connections. For globe valves, the net loss chargeable to the valves was found by subtracting from the measured pressure drop the computed loss in a straight pipe equal in length to the distance between the manometer connections. To determine the loss in the test piping for the angle valves the test valve was replaced by an elbow and the pressure drop across this system at various rates of flow was measured. Then, the pressure loss in one elbow was calculated and this value was subtracted from the measured pressure drop in the test piping with the two elbows in place. This latter figure was, in turn, deducted from the total measured loss caused by the test valve and piping to give the net loss attributable to the test valve only. Thus, the loss charged to the angle valves was not only the loss within the valves themselves, but also the loss caused by a 90-degree change in the direction of flow.

THE EQUIVALENT LENGTH:

Flow studies such as have been described herein would be rather limited in usefulness if the results could be applied only to valves identical to those tested. To increase the value of the test results, it is desirable to employ a method by means of which the results of tests on one valve may be extended so that the performance of other similar valves may be predicted.

Expressing the pressure drop in the valve in terms of the length of straight pipe necessary for equivalent resistance to flow, affords a convenient but only approximate means of accomplishing this. For steady flow in straight pipes, the pressure drop, if small, may be expressed by the equation

$$\Delta P_p = \frac{fV^2}{2g} \rho \left(\frac{L}{D}\right),$$

in which

ΔP_p = pressure drop in the pipe — pound per square inch.
ρ = density of fluid — pound per cubic foot.
V = mean velocity of flow — feet per second.
L = length of pipe — feet.
D = ID of pipe — feet.
f = friction factor — dimensionless.

The friction factor f, for pipes of the same "relative" roughness, is a function of the Reynolds number

$$f = \varphi\,(Re).$$

At high Reynolds numbers the friction factor becomes very nearly constant, *i.e.*, independent of the value of the Reynolds number. Therefore, the pressure drop formula may be written

$$\Delta P_p = C_p\,\rho\,V^2\left(\frac{L}{D}\right)$$

where C_p is a constant.

It can be shown by dimensional analysis that the pressure drop in a valve is given by

$$\Delta P_v = \rho\,\frac{V^2}{g}\,\theta\left[Re,\ \frac{K_1}{D_1}\ ,\ \frac{K_2}{D_2}\ ,\ \ldots\ldots\right]$$

where θ indicates a function of the quantities enclosed in the brackets. The ratios, K_1/D_1, K_2/D_2, . . . describe the shape of the valve. If a series of valves are geometrically similar in shape, and if the "relative" roughnesses of the surfaces are identical, then $\theta\left[\frac{K_1}{D_1}\ ,\ \frac{K_2}{D_2}\ \ldots\right] = 1$, and

$$\Delta P_v = \rho\,\frac{V^2}{g}\,\theta\,(Re)$$

The previously described tests on 6-inch valves indicate that $\theta(Re)$ is very nearly constant.

Thus, if C_v is a constant,

$$\Delta P_v = C_v\,\rho\,V^2, \text{ approximately.}$$

The value of C_v must be determined experimentally.

In order to determine the length of straight pipe which will give a pressure drop equal to that in the test valve ΔP_v and ΔP_p are equated :

$$C_v\,\rho\,V^2 = C_p\,\rho\,V^2\left(\frac{Le}{D}\right).$$

Therefore,

$$Le = D\,\frac{C_v}{C_p}\ , \text{ in which}$$

Le is the equivalent length.

From this equation is is evident that the equivalent length divided by the pipe diameter is a constant for any series of geometrically similar valves. The following table gives the resistance of some typical valves in terms of the equivalent length of straight pipe. The values given for the 6-inch valves were determined by test; those given for the other sizes were calculated. Since different sizes of the same design of commercial valves are not truly geometrically similar, the values given for sizes other than 6-inch must be considered as approximations only.

RESISTANCE OF VALVES TO THE FLOW OF STEAM IN EQUIVALENT LENGTH OF PIPE—FEET

Valves Tested	*Pressure Admitted Above or Under Disc*	*Nominal Valve Size—Inches* 4	5	6	7	8	9	10
Globe Valve, Improved Design	Above..	40	50	60	69	79	90	102
	Under..	48	61	73	84	96	109	124
Globe Valve, Original Design..	Above..	54	69	81	93	108	122	137
	Under..	64	80	96	111	127	144	163
Angle Valve, Improved Design	Above..	41	52	62	71	82	93	105
	Under..	33	42	50	58	66	75	85
Angle Valve, Original Design..	Above..	45	57	68	78	90	102	115
	Under..	44	55	66	76	87	99	112
Horizontal Swing Check Valve		13	17	20		26		34
Gate Valve		2.8	3.3	4		5.3		6.8

Note: The above values of equivalent lengths were calculated for pipes having the following dimensions:

Nominal Size—Inches	4	5	6	7	8	9	10
Outside Diameter—Inches	4.500	5.563	6.625	7.625	8.625	9.625	10.75
Nominal Wall Thickness—Inches	.337	.375	.432	.500	.500	.500	.593

CONCLUSION:

The table of equivalent lengths and the charts indicate that the resistance offered to flow by the globe and angle valves tested was materially reduced by the removal of one of the internal reinforcing struts. The table also indicates definitely that, if conditions permit, gate valves should be used in preference to any type of globe valve.

ACKNOWLEDGMENTS:

The writers wish to express their appreciation for the valuable comments and suggestions given by E. C. Petrie, F. R. Venton, and other members of the Engineering and Research Division, Crane Co., Chicago.

ELECTRICAL INSULATION.

By Jesse B. Lunsford, Member.*

This article, the third in a series of four, presents a mass of data which should be of intense interest to those technically interested in electrical insulation.

Because of the possible demand for reprint copies of this article, it is suggested that orders for copies be placed without delay.

PART III—RESINS (SYNTHETIC AND OTHERWISE).

SPIRITS.

Earlier in this series, the reader was warned as to the ideological character of this discussion. If one would seek only technical data, there are other places to look. If, however, the reader is more concerned with ideas which grow out of facts than he is with the mere recital of facts, he is invited to sit in.

Part I stressed the fact that the "life" of its insulation usually determined the "size" of a dynamo; and that an increase of the one could lead to an increase in the other. Part II emphasized the key position which insulating varnish, as the "living stone" holds in this general scheme of things. Part IV will deal with those other stones (inorganics) which, although having never lived, are considered ageless.

But in this, Part III of the series, it is proposed to deal with a class of materials long believed "possessed of the spirits"; to link them with man's long yearning for immortality, and with his discovery and naming of "*electricity.*" Why? Because to get more horsepower we need to get more "spirit," more of permanence; and resins seem to be our best bet in that direction.

Thus, the central theme running through this whole series is the relationship of "life" to "horsepower." In an absolute *desert,*

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without water anywhere, such ideology might not be true; but near to or on the sea, which is the habitat of Naval dynamos, it could not be otherwise than wholly true.

If one will refer to any dictionary he will observe that "spirit" is defined variously as "the breath of life," "life," "the life principle," "animating or controlling principle," etc., etc. Certain fluids are often referred to as "spirits" because of the old chemistry concept of the disappearance of a liquid at one temperature and its reappearance at another (distillation).

When the author, in Part I, threatened to "enliven" the deadly dry subject of insulation he did not mean to call up any "ghosts," but he does believe that a little of looking backward through the centuries may help us in looking forward with a greater appreciation of the relationship of the past to the future. The class of materials commonly known as "resins" can certainly give us that sweeping view. So, before we take up the present, and attempt to prognosticate the future, let us take a look at the past.

TWENTY-FIVE CENTURIES AGO.

From about 600 B. C. until about the middle of the 16th century A. D., the old philosophers' ideas about material facts converged on what was termed an "animistic" theory, hypothesis, or attempt to explain the "soul or spirit resident in an otherwise lifeless substance." When some one rubbed a piece of *amber* (resin to you) and obtained a spark of electricity, how else could he explain this ghostly phenomenon, this coming to life as it were, of something long since dead and buried?

Ye ancient philosopher could not know that this lump of amber, with so much "spirit" as to respond again, after being buried deep in blue clay for untold centuries of time, had once probably been the life blood of a badly bruised tree or shrub. But he did suspect that it meant fortitude to an unusual degree, and he treasured it as something possessing immortality.

So, it is entirely in keeping with the central theme of this series, "life," and a matter in which the electrical profession can take some pride, that the word "electricity" is derived from the Greek word *elektron* for amber, a resin. *Long live* the resins! And the story of Aladdin's Lamp was a beautiful piece of symbolism, right down this alley.

From about the middle of the sixteenth century until about the middle of the nineteenth century A. D., another philosophy was in vogue. The "animistic" or "spirits" hypothesis was being supplanted by the "fluid" theory or "electric fluid," and this concept is responsible for the common reference to electricity, even today, as the "juice."

For a brief spell (1820 to 1884) we had an "etheral" theory to explain matter. But from then until now, we have explained things according to the *electronic* theory. So, it can be said that one resin insulator (*electron,* Greek for *amber*) made quite a record, ideologically speaking.

But amber is not the only resin which responds spiritedly to rubbing: most resins would. A substance would, of necessity, have to be a good insulator in order to hold an electric charge that way. It might be that this so-called "spirit" of other days could be a measure of the quality of a dielectric today. Who has thought of it in that light, and what would be a good method of test? Merely another question which arises, as one looks backward over time in an effort to scan the future. If the idea seems fantastic, it should be easy to prove it so.

NINETEEN CENTURIES AGO.

The spirit of long-buried trees, the symbolism of a magic lamp, the growth of an industry, and the naming of a modern scientific concept linking matter and energy are not, however, all there is to this picture of the "soul or spirit resident in an otherwise lifeless substance." Amber is not the only resin which has played a long-time part in man's ideology. Man is himself a seeker after "longevity," a dreamer of "immortality," and the mystic quality of various natural resins intrigued him long before he came to know enough about electricity to use resins as "insulation" instead of as "charms" or "incense."

Certain of these other resins, more specifically the vegetable "gum-resins," were brought into some prominence just a little over one thousand nine hundred and forty years ago. Three wise men, following a star, and traveling from afar, arrived at the little town of Bethlehem in Judea with a gift apiece. Considering their mission, those three gifts must have represented the ultimate

in *significance* as of that date. And since our legal reckoning of time, to say nothing of other matters, is based upon the date when these three gifts were offered in homage, it might not be amiss to inquire *what* those three gifts were and *why*.

Of those three gifts, one was the so-called "noble" metal, gold. The other two were of the aforementioned *gum-resins*. You may have a pretty good idea of what the "gold" meant, but do you know what the other two gifts, the "resins," meant?

Before being too cock-sure even about what the "gold" signified, let us consider why gold is classed as a "noble" metal, when brightly cleaned copper is much prettier and a lot more useful. Might not even primitive man, with little data but a yen for symbolizing his ideas, have recognized as the chief virtue of gold, the ageless sheen of its surface, its ability to remain *untarnished* despite its association with other baser elements and conditions? In short, did not "gold" suggest to the ancients a quality of agelessness or longevity which, although lacking the "spirits" quality of the resins, did nonetheless defy the usual course in nature of rotting or rusting away?

Now about these other two gifts, the two gum-resins, Frankincense and Myrrh. They were used as *incense,* something to burn and smell. How could their significance rate two to one over gold? Merely to reply that incense burning has had a religious significance for a long time and in many widely separated parts of the world is no answer at all from an engineering point of view which seeks to learn *why*. And this discussion is an attempt at an engineering approach, in spite of any appearances to the contrary.

It might have been that the ancients, everywhere, did just happen to like the *smell* of incense. In view, however, of the fact that there are ever so many other substances which might be characterized as more agreeable in odor, such a general choice does not seem likely. Also, why *burn* it to smell of it, why not smell it as one would a rose, without burning?

Pardon a bit of speculation, but, considering the balsamaceous origin of the incense resins, and the probable origin of the "fossil" resins such as *amber* (elektron) or copal, might they not have been, while living sap, blood relatives as it were? And might

not even the ancients, widely separated in time and place as they were, all have noted some characteristics in common between the aromatic resins from a still living tree or shrub and the fossil resins which, upon being released from a tomb deep in the earth where they had lain for eons of time, still showed "spirit" upon being rubbed or handled? From such a "spirit" to "spirituals" is not a far step in speculation.

As ignorant of science as the ancients were, they still had the ability to observe and the power of reasoning to realize that some beings, some objects and some natural forces had "it." They could not define it or predict it, so they endowed it with certain supernatural powers or influence, even though these apparently living forces were seemingly locked up in objects or substances which in themselves were lifeless. Thus, the concept of "spirits," both of good and of evil influence were conjured up; and thus the two forms of respect, "reverence" and "fear," so closely associated therewith in the mind.

Obviously, a spirit which could remain in an object long since dead and buried, simply could not die, even if the object were burned. In fact, if the object were burned, then the spirit might be set free, to the right great benefit (presumably) of the individual or individuals who were close enough, and who were not averse to the idea of a little personal immortality. If to drink the blood of a brave or good person might be presumed to confer some added bravery or goodness to the spirit of the person drinking it, why could not the immortal spirit of a resin set free by burning not take up new residence via one's nasal passages; i. e., a gain of "spirit" by inhalation.

This is not a sermon, despite the historical or "spirits" angle; nor is it a sales talk, for the author has neither "gold" nor "resins" to sell. They were both *sold* centuries ago. The author has only *ideas* to peddle—ideas so old as to be without price—so old even that they might seem new, unless one has looked backward into time.

Does not that "noble" quality, revered in the *gold* of yesterday, suggest to you the search for more "corrosion-resistance" in the metallic structures of today; "stainless steel," for example, and preservative treatments applied the *electron* way or desig-

nated in electronic terms? And as for the *resins*, synthetic and otherwise, it is not only a question of "new lamps for old," but also, *why* and *how* best to *rub*.

KING TUT.

There is yet another of the "fossil resins" which has made history, although in somewhat different fashion; the reference is to asphalt, particularly the natural Rock Asphalt. This fossil "resin" did not make any reputation for "spirit" but the news about its "longevity," its preservative properties, certainly did travel about among the ancients. It has been described as nature's first and best preservative. Let us take a look at its service record.

The word "mummy" is derived from the Persian "mumia," which means asphalt or bitumen. The British Museum's guide book on its Egyptian collections is reported to state that "Bodies from which the intestines have been removed and which have been preserved by being filled with bitumen are quite black and hard, and, practically speaking, *last forever.*" Note particularly the *last forever* part.

King Tut-ankh-amen was neither the first nor the last person who made a stab at having his mortal remains stay on and on. He did, however, succeed in making the headlines, particularly when the fact was brought out that not only was he cured in "resin" but that he also had about a quarter of a million dollars worth of "gold" trappings with him. Minus the balsamaceous or incense resins and portrayed as Osiris, God of the Dead, he evidently did not lean toward the "spirit" side, but as to his body, he certainly did go it whole hog on the "resin-gold" combination.

PERMANENCE, LONGEVITY, AGE-RESISTANCE.

We do know now that none of the *resins* are immortal, and that their vaunted permanence has been a bit overdone; their spirit or fortitude was relative and symbolic, not absolute or total. In fact, the natural resins which gave the Art of Electricity not only its start but also its very name and general direction, just lack the necessary spirit and fortitude for us to ride them much further if we want to get more life and more power in less space.

The purpose of the foregoing historical references to a few of the natural resins has been rather to emphasize the ideology behind man's use of them for a long time back and to point out how this same idea of spirit, or longevity, or age-resistance, or permanence, forms the central theme of Parts I to IV, inclusive, of this series. The present program of improvement in electrical machinery does not contemplate any change in direction away from the ancient thought on "resins"; rather it proposes an intensification of effort in the same direction. The look backward in time is to lend perspective to our forward trek. It is part of our estimate of the situation.

NATURAL RESINS.

The definition of a natural resin is a matter of debate. One very general, incomplete, and somewhat inaccurate definition might read somewhat as follows: "A class of vegetable substances, insoluble in water, soluble in alcohol, and easily softened or melted by heat. Mostly, they are transparent or translucent, of a yellow-brown color, and are sometimes elastic but are more generally friable and hard at ordinary temperatures. They become electrified when rubbed, and may be divided into several classes, as follows:

(*a*) Those which *exude* spontaneously from plants or from incisions in the stem and branches. As found, these are generally mixtures of gum resins and volatile oils. Probably the most familiar examples are turpentine (pine rosin and spirits of turpentine), benzoin, dragon's blood, copaiba, copal, the various "lacs" (origin of lacquer) and the balsamaceous resins both of the ceremonial anointment and incense varieties (Frankincense, Myrrh, etc.).

(*b*) Those which are *extracted* from plants by alcohols; they generally contain definite carbon compounds. Some examples of this group are gum ammonium, angelica root, Indian hemp, cubebs, manna and squill. The author is through with historical references but the reader might be interested in looking up the locale of the two last named.

(*c*) Fossil resins occurring in coal and lignite beds, amber, asphalt, gilsonite, copaline, kauri gum or resin, fossil caoutchouc, etc. Note the previous historical references to this class of resins.

Shellac, the *bug* lac, does not fit in with the general definition given above because this is an exudation from a bug instead of a plant. But the bug does subsist on plant life; i. e., it is a vegetarian, and it is resinous by other standards, so it is believed to be classifiable as a natural resin. This illustrates the difficulty of defining, briefly and satisfactorily, which of the countless sticky vegetable exudates or extracts are resins and which are something else. For example, the dividing line between vegetable resins, vegetable waxes, vegetable gums, vegetable oils, gum-resins, oleoresins, etc., is indistinct at times because it may be more a matter or degree than of kind; a matter of merged identities as it were.

For the purpose of this discussion, however, it is not of any great consequence where the line is drawn. The discussion will not enter into any further technical distinctions between the several varieties, degrees and mechanisms of formation of the dried saps or oils, because such differences do not affect the picture which the author is trying to paint. After being dried, oxidized or polymerized, into a form which can be utilized as solid electrical insulation, all of these vegetable derivatives have the property in common of becoming "excited" or "spirit" infested. Since this characteristic is, by common acceptance (see any dictionary) the *definition* of a resin, in spite of all qualifying statements to the contrary, the author will take the liberty of using the term "natural resin" in that broad sense. If this violates the reader's own definition or classification, he is urged to consider whether such usage does in any way alter the questions raised or the conclusions drawn hereunder; i. e., does it weaken the ideology expressed?

STOP WATER.

Water, the roadway of all life, is likewise the pathway of all death. Given warmth, moisture and time, that which once lived is soon back to the starting point again. Certain of the natural resins in the fossil category (amber, copal, etc.) appeared to the ancients as having successfully defied long burial; while other fossil resins (bitumen, asphalt, etc.) were used to preserve the bodies of dead kings. And gold was the first known metal of

such "noble" quality as to resist all water, except that great disintegrator, "aqua regia."

Thus, whatever the symbolisms employed, and whatever our starting point, we eventually arrive at the practical significance of *resins* as a water resistor or *water stop*. Whether used for the Hanging Gardens of Babylon, for a Pharaoh's tomb, for an Arab's tent, for the calking of wooden ships, or in electrical insulation, the purpose is always the same, i. e., to stop water.

It is, however, a characteristic of all engineering materials and designs, that superiority in one respect is attained at a sacrifice in other directions, particularly where the several characteristics so eagerly sought are "in opposition." This is very true of our problems in insulation, as the following discussion will show.

Rubber (when properly compounded) from the sap of a tree known as Hevea Brasiliensis, is hard to surpass as a means of stopping water. As long as the service operating temperature is very low (below 60 degrees C. in most cases, and not higher than 75 degrees C. in any case) there is no known economic equal of this natural rubber. It stretches mightily, even at 0 degrees C., but decays rapidly at the upper temperature limits just cited.

The fossil resins (asphalt for one example) are not so subject to decay (oxidation) as rubber, but they are all *brittle* at low temperatures (0 degrees C., for example) and *flow* at the high temperatures. They "run out on" one, as it were, if the temperature becomes too great.

If the temperature range must go up to 90 degrees C. to 100 degrees C., a composite material such as varnished cambric becomes the best material to employ, simply because at this point more has been gained in heat resistance than has been lost in moisture resistance. Numerous illustrations might be given along these lines, but it is believed that enough has already been said to emphasize the following points:

(*a*) Some form of natural resin (including fossil resins), or natural gum, or natural gum-resin, or a combination of these, has heretofore been found necessary to "stop water."

(*b*) All such natural organic substances have very low, definite temperature limitations either as to "life" (oxidation) or to "flow" (thermo-plasticity).

(*c*) All are highly "flammable."

Whether the "natural" gum, resin or gum-resin, is derived from the sap of a tree indigenous to Brazil (rubber), from the oil of a nut from a tree of China (Tung or china-wood oil), from the secretions of an insect in India, from the fossilized remains of trees along the Baltic Sea, Zanzibar, Australia, or the East Indies (amber, copal, kaura, etc.), or from the oils (linseed oil) of seeds nearer home (flax seed, cotton seed, etc.) the limitations in regard to temperature are much the same, and differ only in degree. All life being nature's syntheses in aqueous solution, it is to be expected that the temperature at which water ceases to be a liquid at normal atmospheric pressure (100 degrees C.) is the temperature at which life stops and decay begins. Were this not a top ceiling how else could the world have escaped being cluttered up with the debris of former life; how could new life have a chance to succeed the old?

WATER, HEAT, AND FLAME.

If one would rise above that ceiling of 100 degrees C., and particularly if the questions of thermo-plasticity and flame-resistance are to be taken into consideration, it is obviously necessary to select insulating mediums not of natural organic growth (at least not without benefit of considerable further synthesis). Thus, the first mediums which come to mind are the *inorganic stones* such as asbestos, glass, mica, etc., mentioned elsewhere in this series. These are truly heat and flame resistant, because they are in themselves oxidation products and hence stable.

Such inorganic insulations are, however, and as explained heretofore, incapable of being bent unless used in the form of extremely fine fibers or extremely thin sheets or flakes. In either case, such fine sub-division of the inorganic substances introduces many interstices. These inorganics, being silicates, the surfaces of such fibers or flakes can be wetted by water. The finer the fibers, the greater becomes the number of the intersticial spaces and the smaller their cross-section. Capillary action being what it is, the tendency of such a mass of fine fibers to take up water might be said to be directly proportional to the number of such intersticial spaces (capillary tubes or chambers) and inversely proportional to their size. Thus, one does not keep water out by further subdi-

vision of fibers, this merely succeeds in raising the pressure tending to fill the capillaries.

So, in order to "stop water" we attempt to impregnate the inorganic fibers or flakes, i. e., to *coat* over their *surfaces* and to *fill* the *interstices* between them. And this is where our difficulties begin all over again; the only effective seals against water, which are even relatively free from other and more serious faults, are the same "natural" resins, waxes, gums, and gum-resins which were formed in nature with water itself as the vehicle.

Such a combination of organic and inorganic materials will provide moisture resistance to a satisfactory degree, and flame resistance to an intermediate degree, but the heat resistance still is no greater than that of the particular organic substance used to stop water. Thus, it is possible to stand our ground in one direction (stop water) and to gain considerable ground in another direction (stop or slow up flame), but in the third direction (heat resistance) little further ground can be gained in the use of "natural" organic materials, because, as explained previously, nature has set a top limit so as to dispose of its dead and to make room for the new, living.

As the temperature problem is purely relative, it may be the more readily visualized if stated in the following ways:

(*a*) Natural organic insulation can be satisfactory, provided it is always *below* the point where water *does boil.*

(*b*) Inorganic insulation can be satisfactory, provided it is always *above* the point where water *has boiled.*

(*c*) A combining of (*a*) and (*b*) can provide a partially satisfactory one-way crossing of this boiling water *divide,* but it can not be counted on for the way back.

Such a combination does help considerably in flame resistance, for example, and it may even be considered as having a high reserve value in that the inorganic component serves as a conductor *spacer* during and immediately following an emergency, after the organic material has cracked up or charred and hence lost its water-stop properties. This may permit the ship to go on, or to limp home, while the heat is still on, but when the cool-off comes the machinery may require replacement. It does not represent the ideal to be striven for, but it does help a lot.

To express the situation a bit differently, neither electrical machinery nor the insulation used therein can, for maximum results, be kept either always below the great divide (100 degrees C.) nor always above it. While a one-way crossing of that divide has long been possible, and it is a step in the right direction, we have a *two-way* crossing job to meet. We can not expect to find the answer to that job in nature, because it would be, the author believes, a direct contradiction to the natural law of disposing of its dead.

SYNTHETICS.

This brings us down to the main subject of this article. By getting away from the "natural" organics, there automatically fades out of the picture, certain critical temperatures as 100 degrees C., 4 degrees C. (greatest density of water), and 0 degrees C., which are of such significance in respect to water and the *life* which depends upon water in the liquid form as the vehicle both of growth and of decay.

In the field of synthetics there are many limitations, including water and heat and flame. But such limitations are specific to the particular type or class of synthetic plastic employed; there is not the general sameness in their mechanism of formation, and of their disintegration, as there is in the growth and decay of vegetable matter; hence, no such broad generalizations appear permissible. If any broad, general conclusion is in order, it might be expressed somewhat as follows:

Never having lived, in the sense of never having existed as living tissue laid down in water or formed through the media of water, water (as such) may be considered as inherently foreign to them rather than merely a divorced former associate. Water, as such, did not make them and hence water, as such, is not necessarily the mechanism of breakdown, i. e., their relationship to water is incidental rather than fundamental, and the effects of water are specific to a type rather than phenomena general to all.

If temperature is the limitation, it is more likely to be one of thermo-plasticity than of oxidation, of physical deformation (flow) under load rather than actual fiber rupture on bending. Whatever their several limitations may be, and even where they

parallel those of the natural organics in numerical terms, the synthetics may exhibit such other differences in characteristics as to require different means of measurement, and also to offer different solutions to the same problems. The field of choice is newer, less explored, and limited more by our present boundaries of knowledge than by the mere *survival* value of living organisms from which the "natural" organics stem.

Earlier in this article there was given a brief outline as to the "natural" resins, gums and gum resins under discussion. As the dividing line between the naturals and the synthetics is exceedingly hazy in spots, it might be well to list those materials in the latter category which are, or which may be of importance as electrical insulation in naval shipboard practice. As "synthesis" simply means a step by step build-up of new substances out of old parts, i. e., a combining of certain elements to get certain results, a recombination of the same elements in another way to get other results, or a combination of other combinations to obtain a desired result. There is a considerable question as to how much *modification* can be done on a natural product before it can properly be classified as a "synthetic."

Even a substance the like of which is not found in nature and hence is unquestionably classifiable as a synthetic on that score, may be combined with other natural substances in the form of reinforcements, fillers, adulterants, modifiers, etc., so that the resultant mass is neither wholly one thing nor the other. In any case, however, if the essential feature of, or the merit inherent in the build up, is a result of the recombination of elements and is thus an improvement on the "natural" product, it might be listed as a "synthetic."

The "synthetics" which, in one form or another, do now have useful applications in electrical machinery and equipment aboardship, or which might prove useful in the near future, may be listed as follows:

Acetates. (Cellulose-acetate, cellulose-triacetate, vinyl-acetate, etc.)

Alkyds. (Glycerin-phthalic anhydride or "glyptal," alkylated phenols, etc.)

Aceto-Butyrate.

Acrylates. (Methyl acrylate, methyl-methacrylate, etc.)
Ethyl Cellulose.
Halogenated. (Chlorinated naphthalenes, chlorinated Biphenyl, rubber hydrochloride, chlorinated rubber, etc.)
Phenolic. (Phenol-formaldehyde, phenol-furfural, etc.)
Polystyrene.
Rubber. (Chloroprene, Butyl rubber, etc.)
Urea. (Urea-formaldehyde.)
Vinyl. (Vinyl-acetate, vinyl-chloride, polyvinyl chloride, the co-polymers (vinyl chloride-vinyl acetate), and polyvinyl acetal.)

It will be observed that the above list is far from being comprehensive, in that it does not show all the synthetics which might prove useful; nor is such listing to be taken as an assurance that their use is justified. Again, the listing is too general in its possible implications; it follows a loose classification as to types but is not specific as to which of the multitudinous compositions in any one group could be usefully employed or for what purposes. To go still further in qualifying the arrangement, some of the examples listed in each of the eleven main classifications are not even akin as to origin, they might better have been switched around some except that after being so switched, they would still be out of place. For example, let us take rubber-hydrochloride, chlorinated rubber, butyl rubber, and chloroprene; they do not fit well together in any one classification, yet to list each of them separately, without linking any of them together in some fashion, would defeat the author's purposes in the conveyance of some general ideas as to possible service uses. So the groupings shown, indefensible as some of them may be from a chemistry concept, are given with other purposes in mind.

CHARACTERISTICS IN OPPOSITION.

It is not the purpose of this discussion to "sell" synthetics, either singly or collectively; nor to make any odious comparisons one with another. It is against the author's philosophy of materials, including synthetics, that there ever was or ever can be any perfect material, nor even any truly "superior" material. There is no material which of itself is either "good" or "bad" per se. It is relatively good or relatively bad as the particular use to which it is put makes it so.

According to that philosophy, every gain in one direction has its "price," i. e., it is attained at the expense of some other trait. If anything is so nearly perfect in one respect, it just has to have at least some one or more very weak spots somewhere. And the first thing to do when one stumbles upon what seems at first glimpse to be a paragon of virtue is to start looking for that weak spot, or spots; the "price" paid to achieve it. A fertilizer having the highest rating for the rose garden could be expected to have certain drawbacks when applied to the parlor rug. The measure of the "quality" of anything is the degree to which its special abilities are attuned to the specific needs of its particular application; and where its weak points are of slight concern in connection with the specific job it is called upon to do.

THE QUALITY OF SYNTHETICS.

Navy Department specifications covering the proposed *standards of quality,* for certain of the classes of synthetics listed above, are in the course of preparation. It is to be emphasized, however, that such quality standards will not concern themselves with what is a "good" chemical compound, for there just is no such thing. The "standards" will relate to specific "needs" of naval shipboard service, and all questions concerning the naming of a specific chemical composition, the manner of its synthesis, and who makes it, will be irrelevant. The specific virtues desired, and the particular weaknesses permitted, for the service jobs to be done will be so set forth that many creative efforts, under different direction, and starting from various base points, will be invited to shoot at the same service targets.

It is out of keeping with the ideological character of this particular discussion, and impracticable from the standpoint of print space, for the author to take up in detail each of the several classes of synthetics listed hereinbefore. For anyone who really wants to get "technical," full opportunity to do so will be afforded by a reading of the several Navy specifications, as these become available for issue. Every specification requirement must face the opportunity of a free and open discussion with its resultant justification or extinction.

PHENOLICS.

There is, however, one exception as to the detail discussion and data angle. This exception concerns that one group of synthetics which, for the sake of simplicity and brevity, may be referred to as "phenolics." Even the "phenolics" do, however, cover such a large field that it is necessary to subdivide it in order to even partially encompass it. For naval standards purposes the field has been narrowed to two groups of substances, as follows:

Molded phenolics (17P4).
Laminated phenolics (17P5).

Reference to these two specifications will show that each of these two fields have been further subdivided as follows:

Under 17P4—10 types, 5 forms, 3 colors.
Under 17P5—10 types, 2 forms, 4 colors.

And even more subdivisional classifications are being demanded for the satisfactory fulfillment of naval service needs. The number of formulations and combinations that are thus possible and demanded, in spite of all efforts to simplify and standardize on the fewest elements practicable, are indicative of the versatility of this one class of synthetics.

The same general observations might be made with respect to all the other synthetics listed hereinbefore. Each is possessed of some spirit which the other does not have. One of the reasons for mentioning "phenolics" in connection with the symbolism expressed at the beginning of this article is that the basic phenolic "resin" (the phenol-aldehyde) used as the impregnant and binder for most "phenolic" products, has the appearance of *amber,* the first resin to be associated with electricity. And in its early applications, even in a large number today, the amber-like coloring and brilliance lent itself to ornamentation and symbolism as did its early natural counterpart, "amber" ("elektron"). In short, being the first of the man-made ambers to come into wide general competition with the natural product, the age-old symbolism is particularly significant today.

Another reason is that the permanence, the resistance to oxidation, the failure to "rot" or to change shape or character under the normal conditions found in nature, seem to typify that spirit

which the ancients looked up to as being somehow, a bit unusual. If there is anything which better symbolizes not only the thought but even the technique back of producing a "mummy" than do the phenolics, the author has yet to see it. When one takes wood flour, cotton flock, paper, cloth, asbestos fibers or glass threads, and then, after removing the moisture, plasters it thoroughly with this phenolic "resin" and bundles everything up tight for a long rest, he has accomplished a good "hermetic sealing" job against water and/or rotting. Is not such a "mummy" just another version of that very old idea, one that was prevalent even before King Tut? It may not be the *last forever* stuff that was anticipated, but it could be plenty long enough for our purposes.

MOLDED PHENOLICS.

The scope, even of this one subdivision in the field of synthetics, is too broad to permit even a modicum of justice to be done to it in any idea discussion such as this. One little illustration might, however, serve to show how certain advantages which have been taken of "light weight" possibilities in one direction could have their parallel in other fields.

Some years ago the major portion of all connection boxes, fittings, etc., for electric circuits aboardship were of metal; a red-brass composition which was easy to cast and which machined readily. Because of the Arms Limitation Conference of 1922 there ensued a general drive to save weight wherever possible.

One such step led to the development of suitable light aluminum alloys to replace the heavy brass compositions in electrical connection boxes, instrument cases, fittings, etc. This substitution of a light metal for a heavy one resulted in a two-thirds saving in weight; i. e., the weight for a given cross-section dropped from one pound to about five ounces.

Then phenolics came into the picture—a class of materials which seemed to offer great possibilities but which, through lack of adequate control of quality, could not be thoroughly trusted. One article might be splendid, while another one, looking just as fine as the first one, was about as dependable as a hunk of nicely colored mud. The fact, however, that the weight was almost exactly half of that of suitable light metal alloys was most interesting.

This light weight possibility was not the only incentive; a still larger one was the fact that the material itself was such a good electrical insulator that all the insulating bases previously used to prevent electrical grounding could be dispensed with, and even the wiring space could be reduced.

So it came about that the box, case or fitting could then be made even smaller in dimensions than the previous metal ones. All in all, a reduction in weight of 60 per cent could be counted upon. Thus the one pound which had already been reduced to five ounces by going to light metal alloys, now became only three ounces by going away from metal altogether. There was also, as pure gain, the further saving in the space occupied and in the cost to produce, coupled with a better insulated electrical system than before.

While the telling of this is short and easy, the road to its accomplishment was not. That was a road to learning, but not a royal one. It does, however, illustrate the point that possibilities can become probabilities only when the control of quality is assured. There are, no doubt, numerous other synthetics having just as good possibilities; but which can not be translated into achievements until their weaknesses can be determined and controlled.

17P5(INT)—LAMINATED PHENOLICS.

That particular portion of the phenolic field chosen for data purposes concerns the "laminated" group, or such of it as comes within the scope of Navy Specifications 17P5(INT). As this is one class of synthetics which has had a long, varied and useful service record aboardship, and as it is a component of almost every type and class of electrical equipment used, it might be classed almost as a key material. Also, the interest in the Navy's tests in this direction has appeared so genuine, and the cooperation of the manufacturers with the Navy has been so excellent, that a showing of some data might be in keeping with that general interest. Furthermore, the very order of its arrangement, with all trade names and identifying symbols omitted, may assist in spreading some of the "doctrine" which lies back of the preaching, which is the motive behind this series of discussions. Propaganda? Yes! But, in Mr. Webster's famous words, "Just look at it," the tabular data.

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17 P 5 QUALIFICATION TEST DA

BRAND IDENTIFICATION				GENERAL CHARACTERISTICS				PERCENTAGE WATER ABSORPTION								VOLUME RESISTANCE		INSULATION RESISTANCE (MEGOHMS) SPEC 1-B		
				% CURE		DENSITY		SPECIMEN 1-A	SPECIMEN 1-B	SPECIMEN	SPECIMEN	SPECIMEN 2-C	SPECIMEN 3			SPECIMEN 1-A (MEGOHMS)		CONDITION B	CONDITION D	CONDITION
HOLDER	BRAND	COLOR	TEST	ASTM	HOT OIL	SPEC GRAV	WATER DISPL	COND D	COND D	COND D	COND D	COND D	COND D	COND E		COND D	COND E			
		Natural		.19	1.95	1.262	0.5	2.01	1.62	0.46	0.87	0.82	0.56	0.54	0.54	Over	Over	Over	87	85.5
		-		.19	0.65	1.265		1.936	1.930	.488	.888	.884	.864	.884	.899	Over	Over	Over	Over	87
		Black		.22	0.96	1.266	0.6.8	.178	.140	.87	.82	.838	.806	.83	.82	100+	100+	1000	1000	Over
				.24	0.86	1.290	0.5.8	1.967	1.68			.878	.778		.868					
		Natural		.182	0.90	1.274	77.8	1.804	1.780	4.67	.878		1.11			Over	Over	Over	97	Over
		Black																		
		Natural		.16	0.87	1.260	0.6.1	0.91	0.83	.858	.868		1.12			Over	Over	Over	76	Over
		-																		
		-																		
		-		.88	0.73	1.260	.97	.96	.788	.888	.888		.888			Over	Over	Over	Over	Over
		-																		
		Black		.162	0.80	1.298	0.4.7	1.68	1.61	.868	.848	.868	.788	.828	.86	100+	1000	84	92	Over
		-																		
		Natural		.182	.122	1.266	0.6.5	1.17	1.08	.878	.858	.92	.82	.86	.97	Over	Over	Over	Over	Over
		-																		
		Natural		.28	0.81	1.286	76.8	0.71	1.62	.788	.868		.818			100+	1000	100+	1000	Over
		-		.18	0.81	1.286	0.7.6	1.78	1.72				.888							
		-		.88	2.06	1.286						1.87	1.87							
		Black		.27	.96	1.298	0.6													
		-		.892	0.86	1.266		1.888	.772	.868	.888	.788	.788	.868	.838	Over	Over	Over	Over	.88
		-		.88	0.87	1.276		1.81	1.88	.868	.877	.78	.868	.828	.878	Over	Over	1000	1000	1000
		-		.98	0.88	1.288	0.9.4	0.86	1.84	.96	.71	.88	.78			100+	1000	17	88	86
		-																		
		Black		.89	0.89															
		-		.88	0.86	1.276	0.7.6	1.78	1.68		.73	.72				Over	Over	Over	88	Over
		-																		
		-																		
		Gray																		
Values adopted for Spec RP5(CONT) of 1 May 1960								.788	.868	.788	.868	.868	.868			Over	Over	88	86	Over
Max of above values				.88	.86	1.292	0.7.8	0.91	0.87	1.88	1.68	1.97	1.84	1.7	0.9			1000	1000	Over
Min. "				.882	0.84	1.276	0.6.1	0.76	0.777	0.868	0.878	0.78	0.868	.868	.838			78	84	
																		88	89	88

N TEST DATA—NAVY TYPE **P B E** PHENOLIC-MATERIAL LAMINATED

TION RESISTANCE (OHMS) SPEC 1-B						DIELECTRIC STRENGTH (VOLTS PER MIL) SPECIMEN 1-A								DIELECTRIC STRENGTH (KV), SPECIMEN 1-B																IMPACT STRENGTH [redacted] FT-LBS PER INCH SPECIMEN 2-B			FLEXURAL STRENGTH AND DEF					
CONDITION D			CONDITION E			CONDITION A		CONDITION B		CONDITION C		CONDITION D		CONDITION B				CONDITION D				CONDITION E											LBS PER SQUARE INCH			DEFLECTION		
								SHORT TIME		STEP BY STEP		SHORT TIME		STEP BY STEP		SHORT TIME		STEP BY STEP		SHORT TIME		STEP BY STEP														

P B E PHENOLIC-MATERIAL LAMINATED TABLE I

DIELECTRIC STRENGTH (KV), SPECIMEN 1-B																								IMPACT STRENGTH [redacted] Ft-Lbs/In of notch SPECIMEN 2-B					FLEXURAL STRENGTH AND DEFLECTION Average Ult. Str. lbs. per sq. in.										HEAT DISTORTION		COMPRESSIVE STRENGTH Average Ult. Str. lbs. per sq. in.					TENSILE STRENGTH Average Ult. Str. lbs. per sq. in.			
CONDITION B						CONDITION D						CONDITION E						CONDITION B			CONDITION D			CONDITION E			CONDITION B					CONDITION D					CONDITION E											
SHORT TIME			STEP BY STEP			SHORT TIME			STEP BY STEP			SHORT TIME			STEP BY STEP												DEFLECTION (INCH)																					
1	203	403	1	203	403	1	203	403	1	203	403	1	203	403	1	203	403	A	B	C	D	E	A	B	C	D	E	A	B	C	D	E	A	B	C	D	E	TEMP	TIME	CONDITION A	CONDITION C	CONDITION D	CONDITION E				
80	88	87	11	11.7	11.6	11.6	11.6	11.6	11	11	11.2	11.3	11.3	11.3	11	11	11.2	.27	.27	.26	.26	.26	1.40	1.40	1.40	1.40	1.40	1.40	1.40	1.40	1.40	1.40	.405	.405	.405	.405	.405	.405	.405	.405	.405	.405			43500	44500			12500	13500	12500	
97	96	96	97	11.6	11.6	11.6	11.6	11.6	11.6	11.6	11.6	11.6	11.6	11.6	11.6	11.6	11.6	.26	.26	.26	.26	.26	1.40	1.40	1.40	1.40	1.40	1.40	1.40	1.40	1.40	1.40	.405	.405	.405	.405	.405	.405	.405	.405	.405	.405			79500	84000			9500		14000	
86	89	88	88	88	88	86	86	88	16	16.5	15.8	15	15	15	15	15	15	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	.405	.405	.405	.405	.405	.405	.405	.405	.405	.405			41500	43500			12500		14000	
44	48	41	49	54	60	88	87	19.5	17.2	16	15	14	14.5	14.8	15.2	15.7	15.7	.24	.24	.24	.24	.24	.24	.24	.24	.24	.24	.24	.24	.24	.24	.24	.24	.405	.405	.405	.405	.405	.405	.405	.405	.405	.405			40000	42500			12500			
30	42	39	41	40	44	17	17	17	17	17	17	17	17	17	17	17	17	.28	.28	.28	.28	.28	.28	.28	.28	.28	.28	.28	.28	.28	.28	.28	.28	.405	.405	.405	.405	.405	.405	.405	.405	.405	.405			35500	38500			12500	13500	12500	
36	32	29	41	43	41	29.5	29.5	29.5	29.5	29.5	29.5	24.5	24.5	24.5	24.5	24.5	24.5	.28	.28	.28	.28	.28	.28	.28	.28	.28	.28	.28	.28	.28	.28	.28	.28	.405	.405	.405	.405	.405	.405	.405	.405	.405	.405			27500	29500			12500	13500	12500	
100	108	93	97	88	88	84	86	80	18	18	18	18	18	18	18	18	18	.28	.28	.28	.28	.28	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	.405	.405	.405	.405	.405	.405	.405	.405	.405	.405			41500	43500			11864	11894		
107	113	105	76	71	69	69	69	69	69	69	69	69	69	69	69	69	69	.28	.28	.28	.28	.28	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	.405	.405	.405	.405	.405	.405	.405	.405	.405	.405			46366	50750					17189	
86	88	88	89	87	87	18	18	18	18.6	17.6	16.7	11	11	11	11.6	11	10.6	.26	.26	.26	.26	.26	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	.405	.405	.405	.405	.405	.405	.405	.405	.405	.405			37600	44000			19500		17189	
			44	70	63	37	39	51	86	82.5	58	19	155	23	169	167	153	.28	.28	.28	.28	.28	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	.405	.405	.405	.405	.405	.405	.405	.405	.405	.405			34600	40000			12340		11520	
97	91	88	93	84	84	21	19	19	19	19	19	19	19	19	19	19	19	.28	.28	.28	.28	.28	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	.405	.405	.405	.405	.405	.405	.405	.405	.405	.405			38500	42000					14500	
104	102	100	60	51	51	22	21	21	21	21	21	20	20	20	19.5	19.5	19.5	.28	.28	.28	.28	.28	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	.405	.405	.405	.405	.405	.405	.405	.405	.405	.405			41600	45450			11500		13500	
																																												38500	41000			9500			
87	78	85	65	66	65	80	80	80	18	18.9	18	16	16	16.5	16.5	18	20	.28	.28	.28	.28	.28	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	.405	.405	.405	.405	.405	.405	.405	.405	.405	.405			31500	37500			9760			
>50	>50	>50	>50	>50	>50	>50	>50	>50	>10	>10	>10	>10	>10	>10	>7	>7	>7	>10	>10	>10	>10	>10	>15										.07	.07	.07	.07	.07	.07	.07	.07	.07	.07			>30000	17500			>10000	10000	10000	
117	119	115	78	71	69	84	84	87	86			86	86	86	84	84	84	>10	>10	>10	>10	>10	>5										.09	.09	.09	.09	.09	.09	.09	.09	.09	.09			41500	34250			12340	13680	12960	
48	51	55	50	48	46	7	8	9	66	67	65	50	49	46	41	48	44	1.05	1.05	1.05	1.05	1.05	3.5										.115	.115	.115	.115	.115	.115	.115	.115	.115	.115			31500	17800			7760	10980	9750	
																		.88	.884	1.05																													
																		.81	.79	.87																													
																		.28	.35	.42																													

TENSILE STRENGTH					POWER FACTOR				DIELECTRIC CONST				LOSS FACTOR				MODULUS of ELASTICITY					BONDING STRENGTH				
LBS PER SQUARE INCH					1 KC		1 MC		1 KC		1 MC		1 KC		1 MC		LBS PER SQUARE INCH × 10⁶					L.B.S. (using 10 mm ball)				

MIL SPECS 17P4 AND 17P5

17P5 QUALIFICATION TEST DATA-

BRAND IDENTIFICATION				GENERAL CHARACTERISTICS					PERCENTAGE WATER ABSORPTION										VOLUME RESISTANCE				INSULATION RESISTANCE (MEGOHMS) SPEC. I-B								
				% ASH	CURE		DENSITY		PARAFFIN SPECIMEN I-A		PARAFFIN SPECIMEN I-B		1 1/2 HR SPECIMEN		4 HR SPECIMEN		SPECIMEN 2-C		SPECIMEN 3		SPECIMEN I-A (MEGOHMS)				CONDITION B			CONDITION D			CONDITION E		
VENDOR	BRAND	COLOR	TEST		ASBT SOL	HOT OIL	SPEC. GRAV.	FLOW MODEL	FLOW ITY		COND D	COND E	COND D	COND E	COND D	COND E	COND D	COND E	COND D	COND E	COND D	COND E	COND D	COND E	1	1 1/2	3	1	1 1/2	3	1	1 1/2	3
		Natural		.36	7.06		1.388	25.6			2.64	2.52	1.087	1.08	0.878	1.03	0.86	0.78					200+	200+		200+	173							
		Black		.38	6.06		1.398	25.5																									
		Brown		.39	6.86		1.398	26.5			4.96	4.66	4.69	8.86	4.73	8.98	3.95		1.69	1.77													
		Natural		.36	7.71		1.396	25.5			4.91	4.96	8.86	8.86	4.61	8.96	3.91																
		Brown		.36	7.05		1.396	25.5			8.97	8.91	8.99	8.68	1.79	8.96	1.05	1.82															
		Natural		.38	6.82		1.396	27.0			9.66	9.53	8.86	8.88	1.72	8.92			1.46	1.91													
		Natural		.39	7.68		1.396	25.5			8.97	8.97	8.91	8.88	1.91	8.91																	
		Natural		.39	6.15		1.396	25.5			8.94	8.96	1.96	1.93	8.96																		
		Natural		.38	7.73		1.398	25.5			8.93	8.97	8.88	8.88	1.88	1.89																	
		Black																															
		Brown		.36	6.96		1.396	25.5			8.97	8.96	8.96		1.88	1.89																	
		Black		.36	6.45		1.396	25.5			8.97	8.97	1.88	1.89																			
		Natural		.39	7.18		1.396	25.5			8.94	8.96	1.86	1.86																			
		Black																															
Values adopted for Spec. 17P5 dated 1 May 1950.									2.67	2.46	2.63		2.86	2.78	2.64	2.58	2.68	2.77															
Max of data values				.46	7.18		1.406	29.5	.87	4.01	4.9		1.08	1.04	0.99	0.98	1.03																
Min " " "				.36	6.06		1.396	25.0	.817	2.01	2.63		.978	.978	.787	1.00	.888																

EST DATA—NAVY TYPE P B P PHENOLIC-MATERIAL LAMINATED TA

DIELECTRIC STRENGTH (VOLTS PER MIL) SPECIMEN 1-A

DIELECTRIC STRENGTH (KV), SPECIMEN 1-B

IMPACT STRENGTH

FLEXURAL STRENGTH

LBS PER SQUARE INCH

P B P PHENOLIC-MATERIAL LAMINATED TABLE II

| | DIELECTRIC STRENGTH (KV), SPECIMEN 1-B | | | | | | | | | | | | | | | | | | | | | | | | IMPACT STRENGTH Ft. lbs./in. notch SPECIMEN 2-B | | | | | FLEXURAL STRENGTH LBS. PER SQUARE INCH | | | | | | | | | | | | | | | | | | HEAT DEFORMATION DEGREES C | | COMPRESSIVE STRENGTH LBS. PER SQUARE INCH | | | | | TENSILE STRENGTH LBS. PER SQUARE INCH | | | | | |
|---|---|---

SILE STRENGTH				DEFLECTION					POWER FACTOR				DIELECTRIC CONSTANT				LOSS FACTOR				MODULUS of ELASTICITY					BONDING STRENGTH				
				INCH					1 KC		1 MC		1 KC		1 MC		1 KC		1 MC		LBS PER SQUARE INCH X 10^3					LBS (using 10 mm ball)				
			10455																											

MIL-P-17 SPECS 1794 AND 1795

17 P 5 QUALIFICATION TEST DATA—

| BRAND IDENTIFICATION | | | | | GENERAL CHARACTERISTICS | | | | | | | PERCENTAGE WATER ABSORPTION | | | | | | | | | | | | | | | | | | | | | | | | | VOLUME RESISTANCE | | | | INSULATION RESISTANCE (MEGOHMS) SPEC. I-B | | | | | | | | | | | | | | | | | | |

ST DATA—NAVY TYPE P B G PHENOLIC-MATERIAL LAMINATED TAB

	DIELECTRIC STRENGTH (VOLTS PER MIL) SPECIMEN 1-A										DIELECTRIC STRENGTH (KV), SPECIMEN 1-B																IMPACT STRENGTH	FLEXURAL STRENGTH AND DEFLECTIO															
	CONDITION E			CONDITION A	CONDITION B	CONDITION C	CONDITION D	CONDITION E			CONDITION B				CONDITION D				CONDITION E																							
											SHORT TIME		STEP BY STEP		SHORT TIME		STEP BY STEP		SHORT TIME		STEP BY STEP																				

PHENOLIC-MATERIAL LAMINATED

TABLE III

DIELECTRIC STRENGTH (KV), SPECIMEN 1-B																IMPACT STRENGTH FT-LB/IN NOTCH SPECIMEN 1-B					FLEXURAL STRENGTH AND DEFLECTION										HEAT DISTORTION		COMPRESSIVE STRENGTH						TENSILE STRENGTH					
CONDITION B				CONDITION D				CONDITION E													LBS. PER SQUARE INCH Specimen No. A, C					DEFLECTION (INCH)										DEGREES C.		LBS. PER SQUARE INCH Specimen No. A, C						LBS. PER SQUARE INCH Specimen No. A, C					
SHORT TIME		STEP BY STEP		SHORT TIME		STEP BY STEP		SHORT TIME		STEP BY STEP																																	

	POWER FACTOR				DIELECTRIC CONST.				LOSS FACTOR				MODULUS of ELASTICITY				BONDING STRENGTH				
	1 KC		1 MC		1 KC		1 MC		1 KC		1 MC		LBS. PER SQUARE INCH = 10^6 Approximate Mfg. Av.				LBS. (using 10 mm ball) Specimen 1" x 1" x ¼"				
	Dry	Wet	Dry	Wet	Dry	Wet	Dry	Wet	Dry	Wet	Dry	Wet	Condition A	Condition B	Condition C	Condition D	Condition A	Condition B	Condition C	Condition D	Condition E
	64	64	64	64	64	64	64	64					1.28	1.27		1.28	1874	938	1886		
																					945
																	944	701	896		
													1.28	1.27		1.28	980	747	916		
																	1880	741	1560		
																					1150
1980																					
																	1838	1934	1958		
													1.28	1.28		1.27	1875	934	916		
																	> 900	700	1560		
																	1859	1934	1934		
																	944	701	565		

N.D. SPECS 1794 AND 1795

17 P 5 QUALIFICATION TEST

BRAND IDENTIFICATION					GENERAL CHARACTERISTICS						PERCENTAGE WATER ABSORPTION																VOLUME RESISTANCE				INSULATION RESISTANCE (MEGOHMS) SPEC. I-									
					% ASH	CURE		DENSITY			SPECIMEN 1-A		SPECIMEN 1-B		SPECIMEN 2-A		SPECIMEN 2-B		SPECIMEN 2-C		SPECIMEN 3						SPECIMEN 1-A (MEGOHMS)				CONDITION B				CONDITION D					
HOLDER	BRAND	COLOR	TEST	IDENTIFICATION CURE "A" OR "B"		WET OIL	HOT OIL	SPEC GRAV	HARD NESS	POWER FACT	AGED OIL		COND A	COND B	COND C	COND D	COND E	COND A	COND B	COND C	COND D	COND E	COND A	COND B	COND C	COND D	COND E	COND A	COND B	COND C	COND D	COND E	SPEC 1	SPEC 2	SPEC 3	SPEC 4	SPEC 1	SPEC 2	SPEC 3	SPEC 4
		Brown			0.37	2.01		1.256	66		101.5		1.086	1.067	(1.06)	1.068	(1.07)	1.06	(1.06)	1.060	(1.06)	1.060						(Over 200)											
		"			0.44	2.03		1.260			111		1.06	1.06	.96	.965	.971	.964	.975	.966	.967	.962						(Over 200)											
		"			0.47	1.98		1.256	66				.968	1.01	1.07	.997	1.033	.991	.976	.986	.962						(Over 200)											
		Black			0.34	1.82		1.285	73.8				.988	1.00	1.01	(1.06)	.964		1.017									(Over 200)											
		Brown			0.46	2.34		1.278	66.5					.965	.962	.975	.988		.982	.977	.966	.962						(Over 200)											
		Brown			0.43	2.16		1.278			112.5		.941	.946	.969	.952	.964	.96	.96	.97	.96	.96						(Over 200)											
		Brown			0.37	2.16		1.284	66.5					.965	.962	.975	.988		.982	.977	.966	.962						(Over 200)											
		Brown			0.43	2.16		1.284			107		1.06	1.06	.97	.96	.97	.96	.96									(Over 200)											
		Natural			0.46	1.97		1.264			122.5		.980	.971	.971	.962	.971	.962	.971	.962	.962	.962						(Over 200)											
		Brown			0.46	2.31		1.284	66.5				1.075	1.075	1.08	1.07	1.08	1.07	1.08	1.07	1.07	1.07						(Over 200)											
		Brown			0.43	2.31		1.284			107		1.06	1.06	.97	.96	.97	.96	.96									(Over 200)											
		Natural			0.46	1.97		1.264	73.5		122.5		.980	.971	.971	.962	.971	.962	.971	.962	.962	.962						(Over 200)											
		Brown			0.43	2.31		1.284			107		1.06	1.06	.97	.96	.97	.96	.96									(Over 200)											
Values adopted for Spec. 17 P 5 (1951) as of 1 May 1956					0.46	2.16		1.284	66.5		107		1.06	1.06	.97	.96	.97	.96	.96	.96	.96	.96						(Over 200)											
Max. of data values.					0.49	2.34		1.295	73.8		122.5		1.086	1.067	1.08	1.068	1.08	1.07	1.08	1.07	1.07	1.07						(Over 200)											
Min. " "					0.34	1.82		1.256	66		101.5		.968	.962	.962	.962	.962	.962	.962	.962	.962	.962						(Over 200)											

N.D. SPECS 1794 AND 1795

17 P 5 QUALIFICATION TES

BRAND IDENTIFICATION					GENERAL CHARACTERISTICS						PERCENTAGE WATER ABSORPTION																VOLUME RESISTANCE				INSULATION RESIST (MEGOHMS) SPEC.									
					% ASH	CURE		DENSITY			SPECIMEN 1-A		SPECIMEN 1-B		SPECIMEN		SPECIMEN		SPECIMEN 2-C		SPECIMEN 3						SPECIMEN 1-A (MEGOHMS)				CONDITION B				CONDITION D					
HOLDER	BRAND	COLOR	TEST	IDENTIFICATION CURE "A" OR "B"		WET OIL	HOT OIL	SPEC GRAV	HARD NESS	POWER FACT	AGED OIL		COND A	COND B	COND C	COND D	COND E	COND A	COND B	COND C	COND D	COND E	COND A	COND B	COND C	COND D	COND E	COND A	COND B	COND C	COND D	COND E	SPEC 1	SPEC 2	SPEC 3	SPEC 4	SPEC 1	SPEC 2	SPEC 3	SPEC 4
		Natural			.34	1.98		1.238	69		101.5		1.08	1.07			1.07			1.07	1.06	1.06																		
					.48	2.04		1.252			118		1.06	1.07			1.069			1.075	1.070	1.066																		
		Brown			.41	1.96		1.248	71.5				1.03	1.03			1.05			1.05	1.07	1.06																		
					.46	2.17		1.284	73.5		117.5		.978	1.005			.967					.984	1.00	1.03																
		Brown			.37	1.93		1.250	71				.966	1.00			1.067			.966	.968	.968																		
		Brown			.49	2.03		1.260	69.5				1.07	1.08			1.08			1.06	.994	.96	.96	.96																
		Brown			.54	1.79		1.268	69.5				.96	1.03			1.03			1.03	.994	.96	.960	.960																
		"			.38	2.05		1.260			118.5																													
		Natural			.42	2.01		1.262	72				.96	1.00			1.00			1.00	.96	.96																		
		"			.36	2.07		1.265			118.5		.96	1.005			1.03			1.03	1.00	.96																		
		Natural			.37	2.05		1.258	70.5				1.04	1.03			1.03			.96	.96	.96																		
		Natural			.33	1.92		1.256	70				.970	1.00			1.04			1.04	1.07	1.07																		
		Natural			.43	2.03		1.262	69		112.5		.965	1.00			1.03			1.04	1.06	1.06																		
		Black			.37	1.66		1.248	69.5				.97	1.02			1.02			.97	1.05	1.03																		
		Black			.49	1.92		1.262	68.5				.96	1.00			1.02			.96	1.03	.96																		
Values adopted for Spec. 17 P 5 (1951) as of 1 May 1956																																							
Max. of data values.					.60	2.34		1.284	69		118		1.08	1.07			1.08			1.08	1.07	1.07																		
Min. " "					.34	1.79		1.238	66		101.5		.960	.962			.962			.962	.962	.962																		

TION TEST DATA—NAVY TYPE P B H PHENOLIC-MATERIAL LAMINATED

INSULATION RESISTANCE (MEGOHMS) SPEC. I-B						DIELECTRIC STRENGTH (VOLTS PER MIL) SPECIMEN I-A												DIELECTRIC STRENGTH (KV), SPECIMEN I-B																		IMPACT STRENGTH FT. LBS. PER INCH SPECIMEN 2-B				FLEXURAL STRENGTH LBS. PER SQUARE INCH			
CONDITION B		CONDITION D			CONDITION E	CONDITION A		CONDITION B		CONDITION D		CONDITION E						CONDITION B						CONDITION D						CONDITION E													
														SHORT TIME		STEP BY STEP		SHORT TIME		STEP BY STEP		SHORT TIME		STEP BY STEP																	
0.5	1	0.5	0.5	1	0.5	0.5		TIME	STEP OF STEP	TIME	STEP OF STEP	TIME	STEP OF STEP	TIME	STEP OF STEP	TIME	STEP OF STEP	SPACE	1	2	3	SPACE	1	2	3	SPACE	1	2	3	SPACE	1	2	3											
1000	69	801	546				177	126	145	116				44	36			14	11.5	8.8	6.6	6.4	6.1	14	14	13.5																	
1000	51	6.3	3.6				146	106	108	106				156	103			19	18	18	3.0	3.0	2.8	3.5	3.4	3.4																	
85	15	1.9	1.5	.7	.5	1.0	80	116	120	118				73	53			16	13	8.4	8.3	8.1	6.9	6.5	5.5	5.1	5.3	4.5	3.5	3.2	4.4	3.9	4.3	4.3	3.8								
85	15	.15	.15	.11	.15	.16	196	146	148	121				104	58				4.3	4.3	3.5	3.5	3.8	4	2.5	2.5	2.1	1.5	1.5	1.8	2.3	2.5	1.5	1.4	1.5							
175	1.0	.8	.76	1.11	.78	1.17	126	126	124	133				82	69			11	10.3	8.9	6.5	6.9	6.6	6.0	4.0	4.1	4.1	5.7	4.8	4	4.6	1.8	1.4	5.8	4.8							
950	8.8	1.7	1.0	1.8	1.8	1.5	169	141	157	159				99	76			12	6	2.3	1.6	1.6	1.4	2.5	2.5	2.2	2.2	2.2	2	2.1	2.5	2.3	2.5	2.2							
1000	1.9	.72	1.1				233	194	246	209				99	68			10	9.8	8.0	6.8	6.7	6.6	4.0	3.5	2.3	2.4	1.6	1.3													
1000	6.5	1.8	9.1	.4	.6	1.1	215	146	236	177				174	109				9	8.2	10.5	9.3	11	9.8	6.1	6.1	6.4	3.5	3.1	4.4	5.5	5.5	4.5	7.3	7.5	8.1						
1000	1.3	1.6	2.0	1.6	1.6	2.0	240	187	258	163				16	17			11	10.8	12.8	7	6.8	7.5	6.1	5.6	2.7	2.6	2.9	6.3	4.5	4.9	4.8	4.1	5.0							
1000	1.1	.57	.18	.82	.82	.84	249	112	138	89				95	85			136	11	6.3	6	5.8	4.1	2.8	4.4	2.7	3.0	2.5	3.3	2.5	2.1	2.0	4.4	4.1	5.0							
.81	.81	.81	.81	.81	.81	.81	205	110	165	118				96	96			10			9.5			6.5			6.0			8.0			6.5								
91	8.9	9.6	6.6	6.6	6.0	8.0	267	194	246	207				194	140			14			8.5			7.0			8.0			6.8			5.3								
8.6	8.8	8.15	8.8	8.8	8.6	8.6	199	112	165	116				95	95			6			9.5			6.6			6.0			8.0			6.5								

ATION TEST DATA—NAVY TYPE P B M PHENOLIC-MATERIAL LAMINATED

INSULATION RESISTANCE (MEGOHMS) SPEC. I-B						DIELECTRIC STRENGTH (VOLTS PER MIL) SPECIMEN I-A												DIELECTRIC STRENGTH (KV), SPECIMEN I-B																		IMPACT STRENGTH FT. LBS. PER INCH SPECIMEN 2-B				FLEXURAL STRENGTH LBS. PER SQUARE INCH			
CONDITION B		CONDITION D			CONDITION E	CONDITION A		CONDITION B		CONDITION D		CONDITION E						CONDITION B						CONDITION D						CONDITION E													
														SHORT TIME		STEP BY STEP		SHORT TIME		STEP BY STEP		SHORT TIME		STEP BY STEP																	
0.5	1	0.5	0.5	1	0.5	0.5		TIME	STEP OF STEP	TIME	STEP OF STEP	TIME	STEP OF STEP	TIME	STEP OF STEP	TIME	STEP OF STEP	SPACE	1	2	3	SPACE	1	2	3	SPACE	1	2	3	SPACE	1	2	3											
1000	69	17	18				209	169	199					115	119			15	13	11	6.7	6.7	6.4	13	11.5	11.5	11	11.6	10.1													
1000	58	8.7	7.0				246	146	197	198				200	117			6.0	6.0	6.9	7.0	10.0	10.0	13	14.5	14.0	9.2	6.1	8.4													
1000	1000	1000					136	116	141					136	116			10.0	10.5	10.5	6.8	6.9	5.5	8	5.7	5.5	3.1	3.3	2.5													
1000	1000		1.5				169	166	210					166	200						6.2						1.5													
1000	1000	36	2.65	2.55			182	108	162					193	136			10	9.1	7.4	5.4	5.4	5.5	10	6.5	4.8	3.8	3.5	3.8													
1000	1000	1.3	1.5	1.3			199	109	162					91	51			6.0	6.4	5.3	3.6	3.8	3.5	6.0	3.5	2.6	1.8	2.1	1.7													
1000	1000	1.6	1.0	1.6			214	120	201					204	143			6.6	7.0	7.1	4.7	5.7	5.4	6.0	3.5	2.6	1.8	2.1	1.7													
1000	1000	1.01	1.01	1.01			1.04	1.04	1.04					1.07	1.06			6.0	5.5	5.3	5.3	5.4	5.3	5	5	5	5	5	5													
1000	1000	1.04	1.04	1.04			1.04	1.04	1.04					1.04	1.04			6.0	5.5	5.3	5.3	5.4	5.3	5	5	5	5	5	5													
1000	1000	1.8	1.6	1.8			1.04	1.04	1.04					1.04	1.04			6.0	5.5	5.3	5.3	5.4	5.3	5	5	5	5	5	5													
	1000	1000	1000	1000			1.04	1.04	1.04					1.04	1.04			6.0	5.5	5.3	5.3	5.4	5.3	5	5	5	5	5	5													
	1000	1000	1000	1000			1.04	1.04	1.04					1.04	1.04			6.0	5.5	5.3	5.3	5.4	5.3	5	5	5	5	5	5													

PHENOLIC-MATERIAL LAMINATED

TABLE IV

DIELECTRIC STRENGTH (KV), SPECIMEN 1-8

IMPACT STRENGTH

FLEXURAL STRENGTH AND DEFLECTION

HEAT DISTORTION

COMPRESSIVE STRENGTH

TENSILE STRENGTH

PHENOLIC-MATERIAL LAMINATED

TABLE V

DIELECTRIC STRENGTH (KV), SPECIMEN 1-8

IMPACT STRENGTH

FLEXURAL STRENGTH

HEAT DISTORTION

COMPRESSIVE STRENGTH

TENSILE STRENGTH

| | | |

M.O. SPECS 1794 AND 1795

17 P 5 QUALIFICATION TEST DA

BRAND IDENTIFICATION				GENERAL CHARACTERISTICS					PERCENTAGE WATER ABSORPTION						VOLUME RESISTANCE		INSULATION RESISTANCE MEGOHMS SPEC. 1-B		
				% CURE		DENSITY			SPECIMEN 1-A	SPECIMEN 1-B	SPECIMEN 1-C	SPECIMEN 2-A	SPECIMEN 2-C	SPECIMEN 3	SPECIMEN 1-A	SPECIMEN 1-B	CONDITION B	CONDITION D	CONDITION

N TEST DATA—NAVY TYPE F B E PHENOLIC-MATERIAL LAMINATED

N TEST DATA—NAVY TYPE F B G PHENOLIC-MATERIAL LAMINATED

PHENOLIC-MATERIAL LAMINATED

TABLE VI

STRENGTH (KV), SPECIMEN 1-B												IMPACT STRENGTH [redacted] FT. LBS. PER INCH SPECIMEN 2-B					FLEXURAL STRENGTH AND DEFLECTION										HEAT DISTORTION		COMPRESSIVE STRENGTH					TENSILE STRENGTH					PLAS
CONDITION D						CONDITION E											LBS. PER SQUARE INCH					DEFLECTION (INCH)					DEGREES C		Specimens No. 2-C LBS. PER SQUARE INCH					Specimens No. 2 LBS. PER SQUARE INCH (Cross-Section 1 Sq. In.)					
SHORT TIME			STEP BY STEP			SHORT TIME			STEP BY STEP																													
1	2	3	4	5	6	1	2	3	4	5	6	A	B	C	D	E	A	B	C	D	E	A	B	C	D	E	TOP	SIDE	CONDITION A	CONDITION B	CONDITION C	CONDITION D	CONDITION E	CONDITION A	CONDITION B	CONDITION C	CONDITION D	CONDITION E	
70	70	68	68	67	67	68	70	68	68	67	67																		40800	31000				8540	8390				
																													39600	30000				7150	8890				
																																		7340	7280				
87	86	86	10	10	10	10	86	86	86	86	86																		44670	29000				15930	10	12890			
																																		19834		10288			
86	86	87	87	87	87	12	12	21	16	14	21																		44490	26000				19960	15990				
14	61	34	14	88	86	86	86	89	16	17	86																		43900	26500				12500		11000			
																																		10500		10000			
69	70	6	83	84	90	87	84	33	86	84																			46000	29000				14500	9000	14500			
																																		10000		9000			
83	2	28	27	28	85	89	81	91	89	86																			39200	27000				6960		8940			
28	89	61	89	89	85	88	89	89	89	84																			41030	28700				13140	10050				
11	12	61	66	61	6	84	88	28	28	28																			41817	28985						10965			
66	67	33	28	98	88	88	87	88	91	83																			34890	24500						9300			
68	69	66	61	28	26	26	26	68	96	86																			38730	27890				11610		10989			
60	61	30	89	89	6	63	61	91	93	91																			35646	23890				14990		12750			
69	69	80	80	93	90		91	89	80	88																			33300	21600				10800		10100			
36			30			36			30																				31600	21500				7000	7000	7000			
96			36			86			86																				25646	21000				14900	14890	14890			
36			36			36			86																				34600	21600				6960	8960	8960			

PHENOLIC-MATERIAL LAMINATED

TABLE VII

STRENGTH (KV), SPECIMEN 1-B												IMPACT STRENGTH [redacted] FT. LBS. PER INCH SPECIMEN 2-B					FLEXURAL STRENGTH AND DEFLECTION										HEAT DISTORTION		COMPRESSIVE STRENGTH					TENSILE STRENGTH					R
CONDITION D						CONDITION E											LBS. PER SQUARE INCH					DEFLECTION (INCH)					DEGREES C		Specimens No. 2-C LBS. PER SQUARE INCH					Specimens No. 2 LBS. PER SQUARE INCH (Cross-Section 1 Sq. In.)					
SHORT TIME			STEP BY STEP			SHORT TIME			STEP BY STEP																													
1	2	3	4	5	6	1	2	3	4	5	6	A	B	C	D	E	A	B	C	D	E	A	B	C	D	E	TOP	SIDE	CONDITION A	CONDITION B	CONDITION C	CONDITION D	CONDITION E	CONDITION A	CONDITION B	CONDITION C	CONDITION D	CONDITION E	
110	18.8	18.9	66	61	59	76	68	69	66	65	53																		46890	30200				6896	7790				
																																		8160	8960				
60	102	110	66	61	60	96	105	105	66	66	66																		42133	21726				10660		9650			
		88	64	86	66				93	97	117																		40890	23600						12160			
66	68	66	96	96	91	80	63	61	96	96	68																		33500	19700				13480	10120				
66	70	76	62	62	62	76	78	68	86	86	63																		36890	20600						11940			
86	16	16	63	66	66	12	16	108	60	66	66																		34635	20890				8390	10866				
11	6	86	66	66	30	66	61	69	96	96	96																		37533	23803						11890			
72	69	69	96	96	96	86	93	96	93	91	99																		34690	20600						10650			
9	6	6	66	66	66	69	69	61	66	66	66																		31600	20200				11500		10760			
6	66	63	96	96	96	96	96	96	96	96	96																		33100	20733				11640		10854			
96	66	99	66	96	96	86	96	96	99	96	93																		34890	21600				9780		9670			
7	86	88	66	61	66	66	67	66	93	96	96																		31800	20300						10926			
96			96			96			96																				36800	22500				9000	9000	9000			
88			66			100			93																				32000	21000				12000	10800	10760			
96			96			86			86																				34500	20800				6896	7790				

	POWER FACTOR				DIELECTRIC CONST				LOSS FACTOR				MODULUS of ELASTICITY					BONDING STRENGTH				
	1 KC		1 MC		1 KC		1 MC		1 KC		1 MC		LBS PER SQUARE INCH x 10^5					LBS (using 10 mm ball)				
	Dry	Wet	Dry	Wet	Dry	Wet	Dry	Wet	Dry	Wet	Dry	Wet	Condition A	Condition B	Condition C	Condition D	Condition E	Condition A	Condition B	Condition C	Condition D	Condition E
													2.70			2.80						
													2.67	2.63	2.62	2.79	2.78	2100			1950	1890
													2.67	2.64	2.63	2.79	2.78					
																		1963				
																		1703			1777	1749
													2.68	2.66		2.82	2.80	1850			1880	1830
													2.67	2.64		2.81	2.79	1879			1790	1736
																		1706				
																		2046			2060	1950
																		1584			1658	1784
																		2060			1973	2052
																		1960	1810		1910	1800
																		1589			1600	1500
																		2376			2250	2128
																		1492			1600	1500

	POWER FACTOR				DIELECTRIC CONSTANT				LOSS FACTOR				MODULUS of ELASTICITY					BONDING STRENGTH				
	1 KC		1 MC		1 KC		1 MC		1 KC		1 MC		LBS PER SQUARE INCH x 10^5					Specimen PETE LBS (using 10 mm ball)				
	Dry	Wet	Dry	Wet	Dry	Wet	Dry	Wet	Dry	Wet	Dry	Wet	Condition A	Condition B	Condition C	Condition D	Condition E	Condition A	Condition B	Condition C	Condition D	Condition E
													2.68			2.80						
													2.67	2.63	2.62	2.79	2.78	2125	2002		2157	2057
													2.67	2.64	2.63	2.81	2.80					
																		2057			2104	2104
																		1790			1845	1960
																		2025			2100	2000
																		2132	2150		2200	1940
																		1850			1800	1750
																		2075			2150	2050
																		1780			1625	1540

N.D. SPECS 1794 AND 1795

17 P 5 QUALIFICATION TEST DA

BRAND IDENTIFICATION					GENERAL CHARACTERISTICS						PERCENTAGE WATER ABSORPTION															VOLUME RESISTANCE					INSULATION RESISTANCE (MEGOHMS) SPEC. 1-B								
					%	CURE		DENSITY			SPECIMEN 1-A			SPECIMEN 1-B			SPECIMEN 2-A			SPECIMEN 2-B			SPECIMEN C			SPECIMEN 3			SPECIMEN 1-A (MEGOHMS)					CONDITION B			CONDITION D			CONDITION
HOLDER	BRAND	COLOR	TEST	IDENTIFICATION SPEC. 1-A	ASH	INSIT SOL.	HOT OIL	SPEC. GRAV.	NORM. MOLD	SHRINKAGE	COND A	COND B	COND C	COND A	COND B	COND C	COND A	COND B	COND C	COND A	COND B	COND C	COND A	COND B	COND C	COND A	COND B	COND C	COND A	COND B	COND C	COND A	COND B	COND C	1	2.5	4.5	1	2.5	4.5	1	2.5
		Brown			17.15	0.43		1.564	92			0.54	1.98		0.58	1.43		(4.47)	(5.)		0.22	1.40																	
					17.28	1.21		1.565	75		1.11	2.10	2.85		2.08	1.89		1.66	2.11		0.51	1.38					2000	110		230			21	16	13	11	.82	.60	.6	.46
		Brown			16.68	1.68		1.579	92			1.38	2.54		1.98	2.06		1.82									1600	2000		1100			1000	1000	1000	32	26	24	17	13
		"			16.82	1.71		1.575				2.02	2.64		2.06	2.56		2.22	2.81		1.61	2.99											2000	2000	2000	1.8	1.3	1.4	2.7	2.7
		"			16.67	2.17		1.578		0.2	1.91	2.60	2.96	1.79	2.47	2.60	2.24	2.66	2.84	1.87	2.41											1.97	1.64	1.49	1.2	.94	.84	1.5	1.24
		"			17.19	2.16		1.564	76.5			1.62	1.94		.82	.60		1.37									2000	2000		520			2000	2000	2000	170	98	102	80	69
		"			16.96	2.21		1.60		10.2	.88	.98	.80	.96	.84		.98	.88		.88	.98											2000	2000	2000	64	44	40	28	26
		"			16.74	2.20		1.60			1.86	2.01	2.51		1.82	1.87		1.82									1800	2000		780			1800	2000	2000	64	44	40	28	26
		"			16.44	2.23		1.60	2.17		1.82	2.06	(2.)		1.93	1.98		(1.5)									2000	2000		(2)			1.8	.86	.82	1.1	.72	.89	.92	.86
Values adopted for Spec 17P5 (1951) as 1 May 1940											1.4	2.2	2.6		2.0	1.6		1.2	1.6							>	2000		800			200			1.1			.84	
Max of data values					16.8	2.60		1.594	92.		1.11	2.20	2.85		2.08	1.89		1.82	2.11		0.51	1.38					2000			1100			2000			170			1.8	
Min " " "					16.15	1.70		1.560	74.5		.98	.98	.98		.98	.92		.98	.88		.88	.98					1800	2000		780			1.8			1.1			.84	

N.D. SPECS 1794 AND 1795

17 P 5 QUALIFICATION TEST DA

BRAND IDENTIFICATION					GENERAL CHARACTERISTICS						PERCENTAGE WATER ABSORPTION															VOLUME RESISTANCE					INSULATION RESISTANCE (MEGOHMS) SPEC. 1-B								
					%	CURE		DENSITY			SPECIMEN 1-A			SPECIMEN 1-B			SPECIMEN 2-A			SPECIMEN 2-B			SPECIMEN C			SPECIMEN 3			SPECIMEN 1-A (MEGOHMS)					CONDITION B			CONDITION D			CONDITION
HOLDER	BRAND	COLOR	TEST	IDENTIFICATION SPEC. 1-A	ASH	INSIT SOL.	HOT OIL	SPEC. GRAV.	NORM. MOLD	SHRINKAGE	COND A	COND B	COND C	COND A	COND B	COND C	COND A	COND B	COND C	COND A	COND B	COND C	COND A	COND B	COND C	COND A	COND B	COND C	COND A	COND B	COND C	COND A	COND B	COND C	1	2.5	4.5	1	2.5	4.5	1	2.5
		Natural			.29	1.85		1.548	87.5		1.93	2.65	2.90		.98						1.24	1.89	1.99		1.28	1.33		.91	.99													
		"			.69	2.03		1.560		1.0	.78	1.46			1.08						.71	1.23	2.36		.67	1.03		.46	.66													
		"			.67	2.16		1.560	96			2.02	2.89		1.66	.98						1.18	1.78			.71	1.07	94	2000		118	4		2000	2000	2000	2.1	1.7	1.6	2.4	2.4
		"			.18	2.06		1.566	89.5		.71	1.80	1.79		.76						.57	1.43	1.66		.46	.80		.27	.92													
		Natural			.28	1.98		1.565		10.6	1.84	2.78			1.84						.86	2.06	2.06		1.28	1.74		.99	1.18													
		"			.09	1.72		1.565		10.6	1.78	2.46	2.83		.92						.82	1.87	1.93		1.66	1.84		.61	.78													
		"			.66	2.06		1.568		10.7	1.80	2.47			.91						.86	2.08	1.87		1.16	1.66		.98	.93													
		"			.35	2.79		1.560	95.5	10.8	.66	2.46	2.80		1.61						.72	1.87	1.86		.99	1.26		.49	.63													
		"			.39	2.06		1.568		10.6	1.94	2.66			1.97						(1.5)	2.18	2.68		1.11	1.94		.66	.88													
Values adopted for Spec 17P5 (1951) as 1 May 1940											1.7	2.15	2.8		1.6						1.25	2.1	2.6		1.5	1.8		(1.25	1.5													
Max of data values					.857	2.79		1.585			1.98	2.65	2.99		1.97						1.71	2.42	2.68		1.66	1.99		.91	1.07													
Min " " "					.067	1.71		1.548			.61	1.46	1.79		.76						.57	1.43	1.66		.46	.80		.27	.92													

N.D. SPECS 1794 AND 1795

17 P 5 QUALIFICATION TEST D

BRAND IDENTIFICATION					GENERAL CHARACTERISTICS						PERCENTAGE WATER ABSORPTION															VOLUME RESISTANCE					INSULATION RESISTANCE (MEGOHMS) SPEC. 1-B								
					%	CURE		DENSITY			SPECIMEN 1-A			SPECIMEN 1-B			SPECIMEN 2-A			SPECIMEN 2-B			SPECIMEN C			SPECIMEN 3			SPECIMEN 1-A (MEGOHMS)					CONDITION B			CONDITION D			CONDI
HOLDER	BRAND	COLOR	TEST	IDENTIFICATION SPEC. 1-A	ASH	INSIT SOL.	HOT OIL	SPEC. GRAV.	NORM. MOLD	SHRINKAGE	COND A	COND B	COND C	COND A	COND B	COND C	COND A	COND B	COND C	COND A	COND B	COND C	COND A	COND B	COND C	COND A	COND B	COND C	COND A	COND B	COND C	COND A	COND B	COND C	1	2.5	4.5	1	2.5	4.5	1	2.5
		Natural			.61	1.83		1.548	87.5	10.5	1.88	2.89			1.88						1.26	2.16	1.99		.79	1.10		.59	.68													
		"			.39	1.66		1.568	95.5		.96	1.63			.86						.88	1.85	1.85		.78	1.06		.38	.53													
		Brown			.78	2.08		1.568	93.5		.78	2.07			.89						1.13	1.91	1.98																		
		Natural			.97	2.08		1.568	92.5		1.68	2.56			1.18						.93	1.84	1.81		1.04	1.62		.39	1.06													
		Natural			.61	2.06		1.570	93.5		.98	2.04	1.68		.96	.89					1.13	2.06	1.96			1.04																
		"			.67	2.07		1.566	93.5		1.68	2.48			1.04						2.06	2.06	2.01																		
		"			.60	1.82		1.566	91.5		2.02	2.82			2.46						1.71	2.66	1.82		2.32	2.89		1.66	1.96													
		Natural			.15	2.39		1.566	93.5		.82	2.68			1.68						1.66	2.86	2.06			1.68																
		Natural			.61	2.0		1.564	95.5		1.86	2.86			1.88						1.13	2.09	2.06			1.74																
		"			.38	2.38		1.558	93.5		.82	2.01	1.83	.84	.92	.89					.74	2.00	1.86			.91																
		Brown			.63	1.82		1.568	89.5		1.09	1.78	1.69		.82	.66										1.19			1000	1000	69			1000	1000	1000	264	84	84	1000	1000	1000
		Natural			.66	1.79		1.565	96.5		1.86	2.86			1.72						1.20	2.02	1.96			1.63																
		"			.62	2.08		1.566	96.5		.82	2.06			2.08						.82	1.66	2.01																		
Values adopted for Spec 17P5 (1951) as 1 May 1940											2.0	2.2			2.0						2.0	2.6	2.4		2.0	2.4		1.1	1.4												
Max of data values					.870	2.60		1.568	95.5		2.02	2.86			2.46						2.06	2.66	2.06		2.32	2.89		1.66	1.96												
Min " " "					.15	1.60		1.548	86.5		.82	1.78			1.10						.74	1.66	1.96		.78	1.06		.38	.53												

N TEST DATA—NAVY TYPE F B H PHENOLIC-MATERIAL LAMINATED

INSULATION RESISTANCE (MEGOHMS) SPEC. I-B						DIELECTRIC STRENGTH (VOLTS PER MIL) SPECIMEN I-A												DIELECTRIC STRENGTH (KV), SPECIMEN I-B												IMPACT STRENGTH [redacted]	FLEXURAL STRENGTH AND DE
CONDITION D			CONDITION E			CONDITION A		CONDITION B		CONDITION D		CONDITION E						CONDITION B				CONDITION D				CONDITION E				SPECIMEN 2-B	LBS. PER SQUARE INCH
																		SHORT TIME		STEP BY STEP		SHORT TIME		STEP BY STEP		SHORT TIME		STEP BY STEP			
1	2.3	4.5	1	2.3	4.5	TIME	STEP BY STEP	TIME	STEP BY STEP	TIME	STEP BY STEP	TIME	STEP BY STEP	TIME	STEP BY STEP	TIME	STEP BY STEP	1	2.3	4.5	1	2.3	4.5	1	2.3	4.5	1	2.3	4.5	A	B	C

PHENOLIC-MATERIAL LAMINATED — TABLE VIII

ELECTRIC STRENGTH (KV), SPECIMEN 1-B													IMPACT STRENGTH [redacted]					FLEXURAL STRENGTH AND DEFLECTION										HEAT DISTORTION		COMPRESSIVE STRENGTH					TENSILE STRENGTH				
		CONDITION D				CONDITION E							SPECIMEN 2-B					LBS. PER SQUARE INCH					DEFLECTION (INCH)					DEGREES C.		LBS. PER SQUARE INCH					LBS. PER SQUARE INCH				
		SHORT TIME		STEP BY STEP		SHORT TIME		STEP BY STEP																															
203	405	1	203	405	1	203	405	1	203	405	1	203	405	A	B	C	D	E	A	B	C	D	E	A	B	C	D	E	TOP	SIDE	A	B	C	D	E	A	B	C	D	E
44	46	46	38	39	34	34	30	30																							26,500	8,000				7,640	7,970			
34	36	36	33	35	36	36	34	34	33	31	31	27	33	33																	24,400	18,000				9,100		18,000		
44	45	46	45	43	39	39	34	46	50	56	46	45	45																		37,600	20,000				11,472				
50	50	48	37	38	40	38	40	33	34	38	34	35	35																		29,400	19,000				11,200		14,900		
46	46	46	35	38	34	35	34	35	33	35	31	30	35																		24,600	17,800				9,480		16,730		
44	43	45	43	42	42	40	40	40	43	43	44	41	38																		20,400	16,200				9,436		7,692		
46	46	44	43	39	36	36	36	46	53	44	43	34	31																		19,800	13,200				8,720		12,900		
48	44	43	43	41	38	38	38	40	47	49	41	34	35																		23,243	14,725				7,644		11,670		
36	39	38	36	33	30	30	27	38	36	31	30	30	30																		17,700	21,093						13,184		
		30			30			30			30																				27,000	16,000				6,900	6,400	6,800		
		46			46			40			44																				24,000	18,000				9,840		14,880		
		48			48			48			30																				30,000	15,000				8,970		7,692		

PHENOLIC-MATERIAL LAMINATED — TABLE IX

ELECTRIC STRENGTH (KV), SPECIMEN 1-B													[redacted]					FLEXURAL STRENGTH AND DEFLECTION										HEAT DISTORTION		COMPRESSIVE STRENGTH					TENSILE STRENGTH				
		CONDITION D				CONDITION E							SPECIMEN 2-A					LBS. PER SQUARE INCH					DEFLECTION (INCH)					DEGREES C.		LBS. PER SQUARE INCH					AVERAGE LBS. PER SQUARE INCH				
		SHORT TIME		STEP BY STEP		SHORT TIME		STEP BY STEP																															
203	405	1	203	405	1	203	405	1	203	405	1	203	405	A	B	C	D	E	A	B	C	D	E	A	B	C	D	E	TOP	SIDE	A	B	C	D	E	A	B	C	D	E
45	46	46	38	38	36	38	36	36	41	33	31	32	36																								7,645	7,985		7,935	7,494
																																			15,100	18,977		18,870	19,680
																																			15,980				
																																			11,975	11,440		10,895	10,410
																																			13,765	13,195		13,625	13,935
																																			5,625			13,500	13,860
																																			11,125	11,635		11,595	10,680
																																			12,920	12,620		12,620	12,860
																																			9,235	9,060		9,610	9,060
																																			7,000	7,000		7,630	7,620
																																			15,160	14,977		14,870	14,860
																																			7,645	7,985		7,935	7,494

PHENOLIC-MATERIAL LAMINATED — TABLE X

ELECTRIC STRENGTH (KV), SPECIMEN 1-B													IMPACT STRENGTH [redacted]					FLEXURAL STRENGTH AND DEFLECTION										HEAT DISTORTION		COMPRESSIVE STRENGTH					TENSILE STRENGTH				
		CONDITION D				CONDITION E							SPECIMEN 2-B					LBS. PER SQUARE INCH					DEFLECTION (INCH)					DEGREES C.		LBS. PER SQUARE INCH					LBS. PER SQUARE INCH				
		SHORT TIME		STEP BY STEP		SHORT TIME		STEP BY STEP																															
203	405	1	203	405	1	203	405	1	203	405	1	203	405	A	B	C	D	E	A	B	C	D	E	A	B	C	D	E	TOP	SIDE	A	B	C	D	E	A	B	C	D	E
44	44	46	43	40	39	35	34																														10400	10,900		10,540	10,670
49	49	50	45	38	46	46	46																														11,165	11,491		11,903	11,969
46	46	50	100	88	53	51	56																														9,620	9,960		9,896	10,306
51	51	46	45	43	46	41	43																														13,930	13,880		13,880	13,870
57	57	48	36	38	51	51	51	30	43	46	36	48	58																								11,940	11,600		12,000	11,950
58	60	50	51	53	51	50	50																														12,040	12,005		12,145	12,040
51	51	46	36	39	44	44	43																														12,085	12,715		12,160	12,765
76	73	70	80	75	54	43	50																														9,715	9,960		9,900	10,076
55	56	40	56	55	36	36	40																														10,540	10,995		11,163	9,940
64	66	60	80	78	48	48	49	56	64	67	46	46	51																								11,850	11,600		11,900	10,860
68	70	56	56	63	67	76	73	68	83	85	60	82	98																								11,860	11,380		11,900	10,780
36	36	36	66	64	33	34	33																														11,710	10,891		10,870	11,891
33	32	40	58	56	32	33	34																														10,976	10,665		11,076	9,976
		36			36																														9,500	9,960		10,000	9,900
		46			47																														13,930	13,880		13,900	13,900
		46			46																														9,620	9,960		9,930	9,970

MODULUS of ELASTICITY

LBS. PER SQUARE INCH x 10^6

BONDING STRENGTH

LBS. (using 10 mm ball)

IMPACT STRENGTH

Ft. Lbs/In. width

MODULUS of ELASTICITY

LBS. PER SQUARE INCH x 10^6

BONDING STRENGTH

LBS. (using 10 mm ball)

MODULUS of ELASTICITY

LBS. PER SQUARE INCH x 10^6

BONDING STRENGTH

LBS. (using 10 mm ball)

It will be observed that the derivation of the quality standards shown in Specification 17P5(INT) of May 1, 1940, is given in full on each data sheet, Tables I to X, inclusive, herein. Note that in this first even partially complete draft of the specifications, the standards set are based on actual test data and not upon personal conversation or negotiation; also, that they skim the cream off the top of the present commercial crop and do not invent new products nor revolutionize existing practices. But remember, this is a first draft and we start out conservatively. What happens afterwards may be interesting.

If it should happen that the several individual manufacturers are able to recognize their own product and should feel somewhat ashamed in spots or if they should get ideas on how to improve it that might be one good way of promoting a higher rate of evolution. And if they should go one step further and the lowest man in each category should feel bad about holding up the general procession then he might plead for permission to do the very thing we are so anxious to have him do, i. e., make a better try at it. And if, by that time, we argue not over the doctrine we have been trying to expound, but rather as to who was its chief exponent, then that might be the best thing yet. With that in mind, the following tabulations are offered:

Table I .....Navy Type PBE	Table VI .....Navy Type FBE
Table II .....Navy Type PBP	Table VII .....Navy Type FBG
Table III .....Navy Type PBG	Table VIII ....Navy Type FBH
Table IV .....Navy Type PBH	Table IX .....Navy Type FBI
Table V .....Navy Type PBM	Table X .....Navy Type FBM

PIPE HANGERS, PARTICULARLY FOR HIGH-TEMPERATURE PIPE LINES.

By Joseph Kaye Wood.*

The author describes the general method of designing the hanger layout for a high-temperature pipe line with particular attention to one type of constant-support pipe hangers. The article is believed to be a contribution of value in covering one detail of the problem of utilization of high-temperature steam.

Pipe lines are supported on seats underneath or are suspended from hangers overhead. Hangers are more universally used than seats because generally pipe lines are located well above the floor which is reserved for large pieces of equipment and walk-ways. This article is concerned only with hangers.

Pipe lines which do not shift position due to thermal expansion or vibration are suspended from the structural steel work overhead by means of simple steel rods hereinafter referred to as "rigid hangers." When the vertical shifts are appreciable flexible hangers either of the "variable-support" type, better known as simple spring hangers, or the "constant-support" type, such as the bell-crank-and-spring and the simple counterweight hangers, are used. When the lateral shifts are appreciable hangers are provided with rollers or their equivalent.

We will first consider a relatively important and heavy pipe line free of either vertical or lateral shifts and suspended by means of "rigid hangers." As these hangers are relatively slender steel rods the pipe cannot be subjected to turning moments which will reverse the stress in the hangers from tension to compression because of the limitation expressed by Euler's law and sometimes because of the existence of linkages at the ends of the rod. However, in order to bring this condition about, the spacing of hangers

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with respect to weight distribution has to be very irregular. Other considerations such as limiting the tensile stress in the rods to a safe value and preventing excessive sag between adjacent hangers really require a more intelligent spacing of the hangers. It would seem to be good engineering, therefore, even though there be no thermal or oscillatory shifts, to design a piping suspension in accordance with the following fundamental conditions:

1. The sum of pipe weight and supporting forces shall be equal to zero.
2. The sum of the moments of the pipe weight and supporting forces about both the x-axis and y-axis shall be equal to zero.

For example, consider the run of piping shown in Figure 1 which for the time being we shall assume to be an "infinitely" rigid beam. The weight forces designated by W_1, W_2, W_3, etc., etc., are the weights of elemental sections applied at the centers of gravity of those sections while the supporting forces exerted by the hangers are designated as H_1, H_2, H_3, etc., etc., and are applied wherever good judgment and convenience of attachment dic-

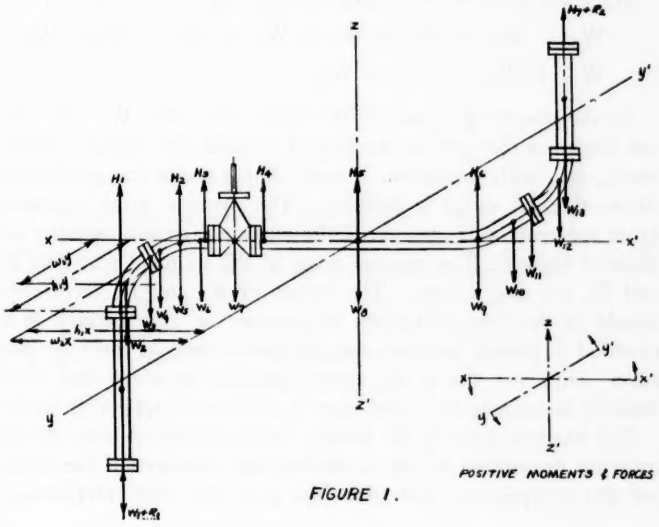

FIGURE 1.

tate. Before going into detail as to what constitutes "good judgment" it might be well to set up the equations expressing the conditions quoted above.

SUM OF MOMENTS ABOUT X-AXIS:

$$-H_1h_1y - H_2h_2y + H_6h_6y + H_7h_7y - R_1r_1y + R_2r_2y = +W_1w_1y + W_2w_2y + W_3w_3y + W_4w_4y + W_5w_5y - W_9w_9y - W_{10}w_{10}y - W_{11}w_{11}y - W_{12}w_{12}y - W_{13}w_{13}y$$

SUM OF MOMENTS ABOUT Y-AXIS:

$$-H_1h_1x - H_2h_2x - H_3h_3x - H_4h_4x + H_6h_6x + H_7h_7x - R_1r_1x + R_2r_2x = +W_1w_1x + W_2w_2x + W_3w_3x + W_4w_4x + W_5w_5x + W_6w_6x + W_7w_7x - W_9w_9x - W_{10}w_{10}x - W_{11}w_{11}x - W_{12}w_{12}x - W_{13}w_{13}x$$

SUM OF FORCES:

$$H_1 + H_2 + H_3 + H_4 + H_5 + H_6 + H_7 + R_1 + R_2 = W_1 + W_2 + W_3 + W_4 + W_5 + W_6 + W_7 + W_8 + W_9 + W_{10} + W_{11} + W_{12} + W_{13}$$

In the preceding equations the values W_1, W_2, W_3, etc., etc., are known as they are the weights of straight pipe, flanges, valves, bends, etc., with insulation, if used, which under the assumption of no thermal shifts is unlikely. The moment arms measured from the centers of gravity of the elemental weight sections are likewise known. The moment arms of the terminal reactions R_1 and R_2 are also known. The values of R_1 and R_2 themselves should in the first calculation be assumed as zero because it is essential to protect terminal pipe line connections as much as possible. However, this is not always possible, in which case these reaction forces may be looked upon as additional supporting forces.

The moment arms of the hanger forces depend to some extent upon the disposition of the structural steel members of the building and of equipment and other pipe lines that offer interference.

The values of H_1, H_2, H_3, etc., etc., are not known and since there are only three equations available for the solution of these unknowns it becomes necessary to assume values for all but one of the hangers, provided the reactions R_1 and R_2 are considered as unknowns. In order to exercise proper judgment in making these assumptions the following simple calculations should be made preliminary to the solution of the equations. Referring to Figure 2 suppose it is desired to assign a value to H_1. We have,

Moment about a—a′

$$\frac{1000 \times 5}{10} = 500 \text{ pounds.}$$

Moment about b—b′

$$\frac{1500 \times 10}{15} = 1000 \text{ pounds.}$$

Weight Rod & Clamp = 25 pounds.

Total = H_1 = 1525 pounds.

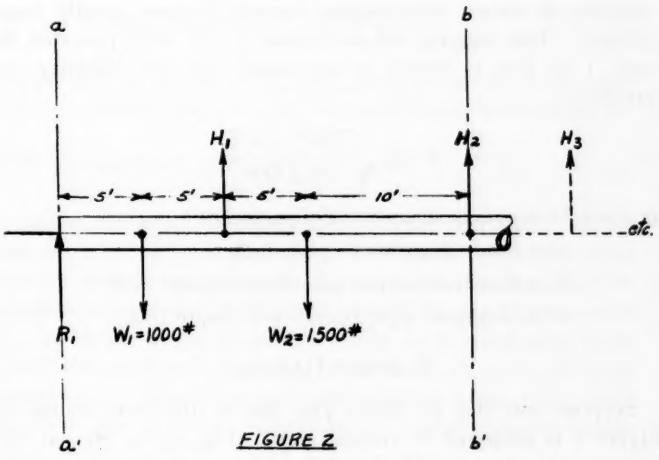

FIGURE 2

In like manner the values of H_2, H_3, etc., can be estimated preliminary to a solution of the main equations. Sometimes it is more convenient to consider one or two of the moment arms as the unknowns. In the case of concentrated weights such as valves or other fittings it is always good practice to place hangers close to the inlet and outlet ends.

The foregoing procedure applies to an elastic as well as to the originally assumed "rigid" pipe line provided the spacing of hangers along the center axis of the pipe line is such as to prevent unsafe sagging. Referring to Figure 3, the center axis of an

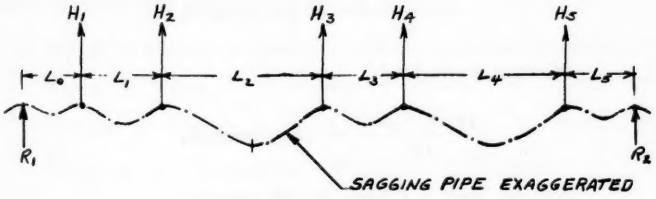

FIGURE 3.

actual elastic pipe line, supported by a number of hangers and two reactions is shown with sagging between hangers greatly exaggerated. This sagging will not exceed a safe limit provided the span, L in feet, is limited in accordance with the following expression:

$$L \leqq \sqrt{\frac{(D^4 - d^4)S}{15.3\,Dw}} \qquad (1)$$

in which D = outside diameter of pipe, inches
d = inside diameter of pipe, inches
S = maximum stress, pounds per square inch
w = weight of pipe, pounds per square foot

Flexible Hangers.

Suppose now that an elastic pipe line of the form shown in Figure 1 is subjected to vertical shifts of ε_1, ε_2, ε_3, etc., at the corresponding hangers H_1, H_2, H_3, etc., etc., due to thermal ex-

pansion. Since this would either buckle the "rigid" hanger rods or induce excessive bending stresses in the pipe some type of flexible hanger must be used instead. With a simple spring hanger the supporting force will vary depending upon the values of the vertical shift ε, the load-deflection rate of the hanger k_s, and the load-deflection rate of the pipe k_p at the point of support. Furthermore, since the percentage variations of the supporting forces of the different hangers will be of different values there will be considerable transfer of supporting effort from hanger to hanger and, most important of all, from hanger to pipe. Transfer of pipe weight from the hangers to the pipe itself is so important because it induces added stresses in the pipe wall which are likely to augment those stresses due to the constraint caused by anchors and guides and to the internal fluid pressure.

The degree of protection which a simple spring hanger affords to a thermally shifting pipe line is measured by the Spring Hanger Factor* K which is equal to $\frac{k_s}{k_s + k_p}$. The transfer of pipe weight from the hanger to the pipe is equal to $\varepsilon k_p K$ or we might write:

$$\text{Weight Transfer} = \varepsilon k_p \frac{k_s}{k_s + k_p} \qquad (2)$$

Setting $\Phi = k_p/k_s$ and $V = 100\, k_s/W$ in which W is the pipe weight, we obtain, per cent Pipe Weight Transferred to Pipe per inch of Vertical Shift $= V \frac{\Phi}{1 + \Phi}$ (3)

The factor V, which is called the per cent Variability of the hanger gives the per cent decrease or increase of supporting effort with each inch of vertical shift. It is obvious therefore that the greatest pipe protection is obtained when V equals or approaches zero. In simple spring hangers V can be made small only by increasing the length of the spring, a value of 20 per cent being obtained for an overall hanger length of about 3 feet. To obtain the fullest pipe protection therefore, a constant-support hanger

* See "The Balanced Support of High-Temperature Pipe Lines," By Joseph Kaye Wood, in "Engineering," December 3rd and 10th, 1937.

which always has a variability factor of zero must be used instead of the simple spring hanger.

Figure 4 shows a family of variability curves which give the per cent of pipe weight transferred to the pipe per inch of vertical shift for different values of the ratio Φ. Usually the value of k_p with respect to k_s is very large which means that the horizontal or flat portions of the curves usually apply. In turn it follows that the per cent of pipe weight transferred to the pipe per inch of vertical shift is practically equal to the variability factor V. It is not safe, however, to assume this as true in all cases, particularly in the case of vertically shifting pipe lines of considerable horizontal length.

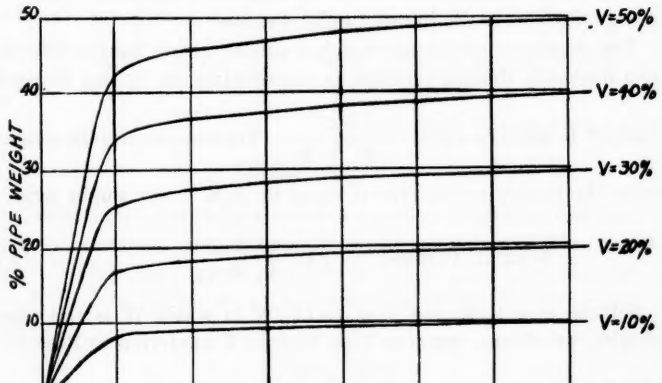

FIGURE 4.

In order to determine the foregoing weight transfers it is necessary to know the amount of vertical shift ε and the load-deflection rate k_p of the pipe at the position of the particular hanger in question. Since adequate methods for determining the vertical and lateral shifts of high-temperature pipe lines have been too well established by other authorities no further space will be given here to that phase of the problem except to say that in the case of "cold-springing" a pipe line the value of ε is reduced by the

amount of the "cold spring" movement, that is, for 50 per cent "cold spring" the value of ε would be reduced 50 per cent. Similarly, methods have been developed for determining the value of k_p, although it is not so essential to know this value at least accurately, because as stated previously, in most cases the flat portions of the curves in Figure 4 apply, in which case the per cent weight transfer per inch vertical shift is equal to the per cent variability.

With a pipe line suspended from simple spring hangers of known variabilities good approximations of the pipe line loadings at the various hanger locations may be made from which the maximum stress due to weight transfer alone may be determined from beam formulas. With a pipe line suspended from constant-support hangers there will not be any pipe line weight loadings and hence no stress induced in the pipe.

Vibrations in Pipe Lines.

Vertical shifts of the pipe may be produced by vibration as well as by thermal expansion in which case the maximum amplitude is equal to ε. Whereas thermal shifts occur with extreme slowness and are practically non-repetitious, vibrational shifts occur with relatively high rapidity which in the case of simple spring hanger suspensions subjects the pipe material to fatigue effects. There is also a further danger so far as vibration is concerned. A simple spring hanger has a natural period of vibration corresponding to its load-deflection rate k_s. These hangers act transversely to the pipe line which also has a natural period of vibration corresponding to its load-deflection rate k_p. A vibration of given frequency is impressed upon the pipe line and if the relation of these three periods is right, resonance will occur. This greatly augments the vertical shift ε with resultant increase of weight transfer which increases the maximum stress in the pipe.

Constant-support hangers on the other hand have no natural period of vibration because the load-deflection rate is equal to zero. Consequently hangers of this type will not enhance the possibility of resonance and if they have a sufficient degree of frictional or other type of hysteresis will act as very effective dampeners to a pipe line tending to vibrate. It is correct, therefore, to consider

constant-support hangers as non-resonant to vertical vibrations but not necessarily to horizontal vibrations. Simple spring hangers are not necessarily resonant to either vertical or horizontal vibrations, although they are very likely to be because the load-deflection rate is greater than zero. Thus simple spring hangers have a natural frequency that is proportional to the square root of the variability factor V and are good for imposed vibrations of frequencies equal to or greater than three to four times the natural frequency.

Installation of Pipe Hangers.

The installation of rigid, simple spring and constant-support hangers may be greatly facilitated by a more exact and simplified layout of the hanger with respect to the elevations of beam and pipe center at the load line which is usually the only information given on the piping drawing. Referring to Figure 5, the "rigid" hanger, consisting of a simple steel rod, is shown. The distance C_b, called the "beam takeout," depends upon the particular type of beam clamp or attachment used. The distance C_p, called the "pipe takeout," depends upon the particular type of pipe clamp or attachment used. Thus the length of the rod R is given by the following expression:

$$R = \text{Diff. Elevations Beam and Pipe} - C_b - C_p. \qquad (4)$$

When flexible hangers, either of the simple spring or constant support types, are used a section of the rod in Figure 5 is cut out as shown in Figure 6 and the lower end of the upper rod R_U and the upper end of the lower rod R_L are attached to the hanger. The length of the section cut out is C_h and is called the "hanger takeout." The elevation of the hanger is the elevation of some well established reference point on the hanger, the location of which with respect to the lower end of the upper rod R_U is given by the dimension N and with respect to the load line by the dimension L.

During operation of the hanger the "hanger takeout" C_h varies because the lower rod moves vertically up or down in accordance with and equal to the resultant thermal shift of the pipe center which according to equation 2 is equal to $\varepsilon\Phi K$ and in the case of 50 per cent "cold-springing" is equal to $\frac{1}{2}\varepsilon\Phi K$.

Since the hanger is usually installed when the pipe is "cold" (62.5 degrees F.) the layout in Figure 6 is based upon that condition with the resultant thermal shift of the pipe straddling the mid point of the total travel range of the hanger. Thus if C is

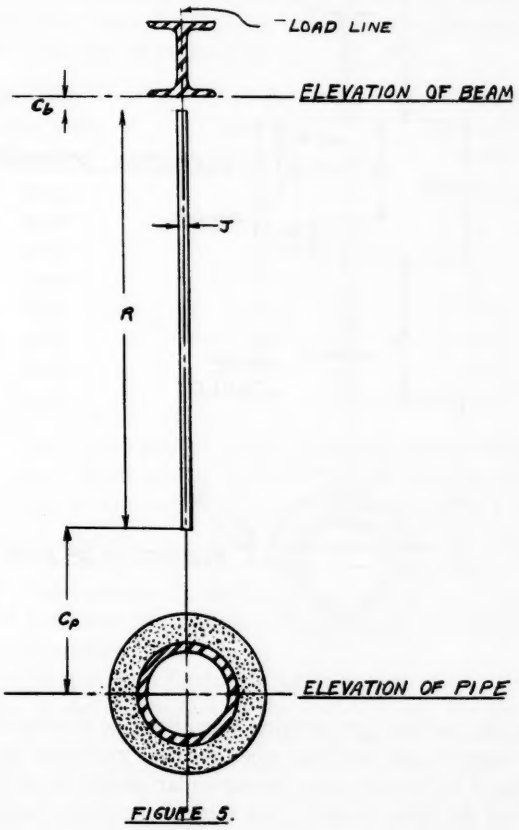

FIGURE 5.

the "hanger takeout" for the mid point of the hanger travel then C_h, the "hanger takeout" for the "cold" position of the pipe, is equal to $C \pm \frac{T_a}{2}$ in which T_a is the resultant thermal shift of

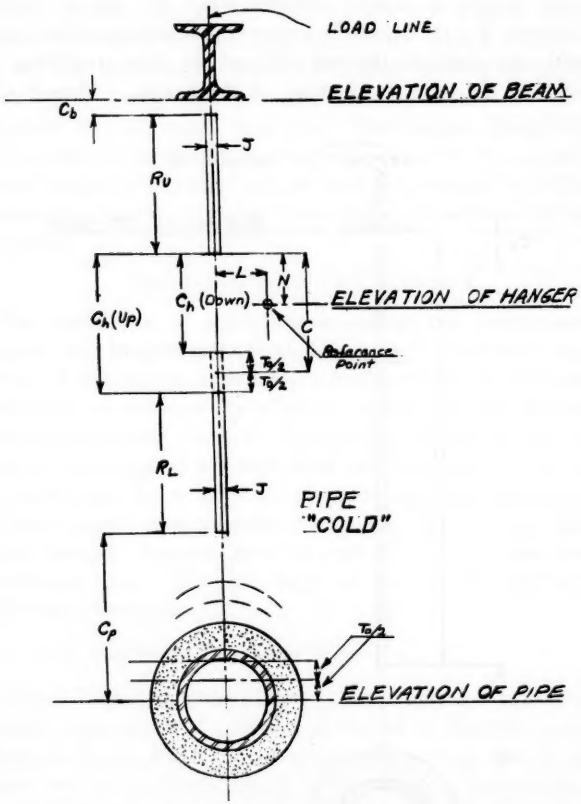

FIGURE 6.

the pipe, the positive sign applying when the pipe movement is up and the negative sign applying when the pipe movement is down. The value of C depends upon the particular design of the hanger.

None of the other factors vary during operation, that is, the elevations of the hanger and beam, the beam and pipe "takeouts," and N remain fixed. With the assumption of a "cold" pipe-line we can therefore express the lengths of the rods as follows:

R_U = Diff. Elevations Beam and Hanger — C_b-N. (5)

R_L = Diff. Elevations Hanger and Pipe — (C_h-N) — C_p (6)

Before discussing specific designs of hangers and their corresponding "takeout" or C values, it may be mentioned in passing that for one well-known manufacturer's I-beam clamp C_b has a constant value of 2 inches and for his adjustable pipe clamp C_p has a value depending upon the pipe weight to be supported. The following values of C_p have been found to give suitable results for this type of pipe clamp.

Max. Pipe Hanger Load Pounds	*Safe Pipe Clamp Sizes Inches*†	*"Pipe Takeout" C_p Inches*
2500	6	6-7/16
3500	12	10
4500	16	13
5500	18	14
7000	20	15¼
8000	24*	17¾
9000	24*	17¾
10,000	24*	17¾

Note: * For 8000 pounds, width of ¾-inch strap = 3 inches.
* For 9000 pounds, width of ¾-inch strap = 3½ inches.
* For 10,000 pounds, width of ¾-inch strap = 4 inches.

† Actual nominal pipe sizes can be equal to or less than these sizes.

In a similar manner C_b and C_p values can be listed for any designs of beam and pipe clamps whether they be standard commercial or tailor-made types.

The reference point for simple spring hangers may conveniently be taken at the intersection of the load line and the top surface of the spring seat. Figure 7 shows the hanger elevation and the "takeout" distance C_h based upon this reference point. In this case the value of the offset L (Figure 6) is equal to zero.

In the bellcrank-and-spring hanger, the reference point is taken at the center of the main pivot about which the bellcrank lever rotates. This is also true of counterweight hangers. Figure 8 shows the hanger elevation, "takeout" C_h, and N for a given make of the bellcrank-and-spring type of hanger based upon this

reference point when using an adapter bracket and when not using one. This bracket converts the two-rod upper suspension into a single-rod upper suspension, it being optional with the layout draftsman as to which one should be used. The value of the nor-

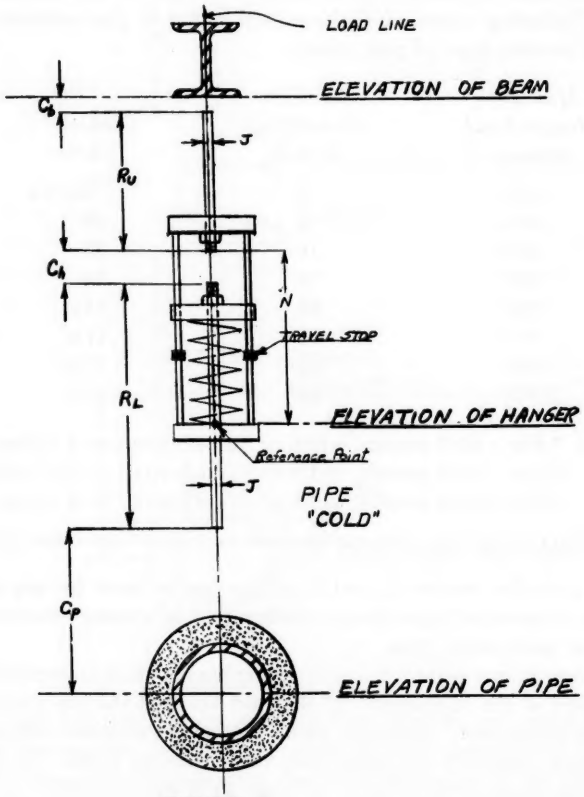

FIGURE 7.

mal "takeout" C for this hanger, which depends upon the total travel of the hanger, is given in Table I. In this same table the values of L, the lateral offset of the main pivot center from the load line, and the value of N also are given.

In passing it should be stated that the overall dimensions of hangers should always be given with reference to the main pivot center which makes it possible to provide the necessary clearances against interferences of the hanger with adjacent equipment.

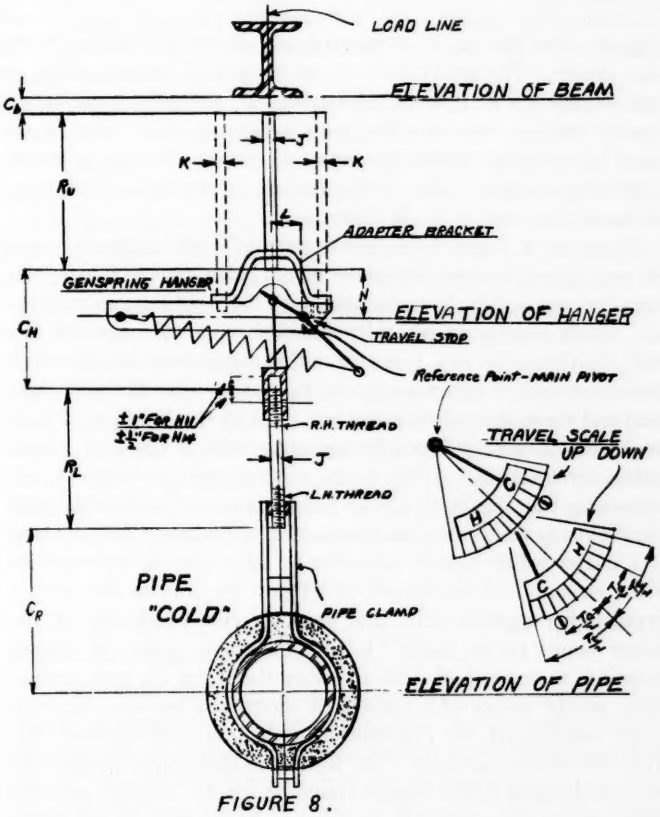

FIGURE 8.

In Figure 8 a travel indicator shows the position of the pipe center at any time. The "cold" and "hot" positions are shown by the "C" and "H" markings on the scale which has 10 divisions. Since the actual travel or shift of the pipe is made to

straddle the mid-point in the total travel T_t of the hanger, the number of divisions along either the "C" or "H" markings at which the position $\frac{1}{2}$ T_a is located from the mid point on the scale is $5\frac{T_t}{T_a}$. The travel indicator is shown diagrammatically in Figure 8 for the cases of upward and downward shifts of the pipe center. The total travel of the hanger T_t should always be greater than the shift of the pipe center T_a, the greater the better. Spring hangers are provided with safety stops so that in the event of a spring failure the pipe will not be released from the supporting hanger. This safety feature is required of all types of hangers by the A. S. A. Code.

Equations 4, 5 and 6 are based upon a "cold" pipe and upon the assumption that the elevations of the beam, pipe and hanger in actual service will be in accordance with the drawings, an assumption which usually is not fully justified. To take care of any such discrepancies two types of adjustments may be provided. First, one end of either an R_U or R_L rod may be threaded right hand and the other end threaded left hand so that when the rod is turned the distance between the engaging units is changed to suit, within certain limits. This is the method used for the R_L rod connecting the Figure 8 spring hanger and the adjustable pipe clamp already mentioned, the limits of adjustment of length being $\pm$ 2 inches which is quite adequate to take care of discrepancies in the elevation of the hanger and pipe. In practice this rod is turned either right or left hand until the travel indicator of the hanger points to the mark "C" on the travel scale. A second method is to secure both ends of either the R_U or R_L rods permanently and by means of a turnbuckle interposed between the ends of two sections of the rod shorten or lengthen the distance between the end connections. The R_U rod in this spring hanger has one end clamped to the hanger frame or adaptor bracket and the other end may be clamped or screwed into a nut of the beam clamp, the threading being right hand. A change of length corresponding to limits of $\pm$ 1 inch may be obtained this way which is quite adequate to take care of discrepancies in the elevations of the beam and hanger.

Having the lengths of the R_U and R_L rods it remains necessary only to determine the diameters of these rods. The diameter of a hanger rod must satisfy two requirements, first, strength, and second, rigidity against bending. The former is governed by A. S. A. practice and the latter by the length. Table II gives the allowable diameters of rods for use with the spring hanger of Figure 8 a practice well within the A. S. A. requirements and giving dependable results.

For a given size of spring unit the product of the supporting force (load) and corresponding total travel in Figure 8 is a constant. Table III gives these products for the large heavy duty hanger and Table IV gives the same for the smaller marine type. Unlike the simple spring hanger this hanger can be adjusted to give any supporting force (load) within a range of about $\pm$ 17 per cent of the Rated Capacity, which is the test load of a particular hanger when the load adjustment is set to normal. In operation the supporting force is always constant throughout the total travel of the hanger.

All sizes of this type of hanger can be used where the lateral shift in any direction from the load line at the pipe center does not exceed a length equal to one inch for every foot difference in elevations of the hanger and pipe. For lateral shifts exceeding this rule separate provision has to be made in the means for attaching the hanger to the structural steel.

Summary.

Summarizing, the step-by-step procedure in the design of a pipe line suspension is as follows:

1. Determine the weight reactions and weight supporting capacities of all hangers for the locations given on the drawings under the assumptions that the sum of all weights and weight supporting forces, including the terminal reactions, is equal to zero and that the sum of the moments of these forces about both the x and y axes is also equal to zero.

2. Check the stresses due to sagging between any two adjacent hangers or between a hanger and a terminal connection.

3. Determine the vertical (ε) and lateral shifts and the load-deflection factor k_p at the points of support.

4. Assuming a given variability factor V for each hanger, let us say 20 per cent, calculate the weight transferred from each hanger to the pipe, taking into account "cold-springing" if any is used.

5. Assuming the transfer weights of paragraph 4 as the only forces acting on the pipe line, determine the maximum stress.

6. Determine the maximum expansion stress due to the restraining action of anchors and guides, but ignore the effect of pipe weight. Combine this with the maximum stress due to the internal fluid pressure to obtain the maximum resultant stress.

7. If the stress in paragraph 5 augments the maximum resultant stress in paragraph 6 to an unsafe degree, substitute constant support hangers for the simple spring hangers. With these hangers there will be no transfer of weight to the pipe and hence no augmentation of the maximum resultant stress in paragraph 6.

8. If the use of bellcrank-and-spring hangers has been decided upon, refer to Table III or IV and select the proper size spring unit and total travel corresponding to the required supporting force of each hanger.

9. Referring to Table I, ascertain the value of the "hanger takeout" C for the mid travel point and from this and the vertical shift of the pipe T_a $(= \varepsilon \Phi K)$ calculate the value of the "hanger takeout" C_h $\left(= C \pm \frac{T_a}{2}\right)$ for the "cold" position of the pipe.

10. From the known values of the pipe, beam, and hanger "takeouts" (C_p, C_b, and C_h, respectively) and the elevations of the pipe, beam, and hanger along with the value of N (Table I), compute the length of the upper and lower hanger rods (R_U and R_L), respectively, for the "cold" position of the pipe.

11. From Table II, select the proper diameters of rods R_U and R_L for the corresponding supporting force of each hanger.

12. From the known elevation of the hanger and the value of the offset L determine if the hanger will have sufficient clearance to prevent interference with adjacent equipment.

TABLE I—BELLCRANK-AND-SPRING HANGER (FIGURE 8).

DIMENSIONS APPLICABLE TO ALL HANGER SIZES.

Heavy Duty Type

Total Travel, Inches	*Load Line "L," Inches*	*"Take-Out" Dimension "C" for Calculating Rod Lengths*	
		Without Bracket, Inches	*With Bracket, Inches*
2	2-5/8	8-5/16	14-5/16
2-1/8	2-13/16	9-3/8	15-3/8
2-1/4	2-15/16	9-5/16	15-5/16
2-3/8	3-1/8	9-1/4	15-1/4
2-1/2	3-5/16	9-3/16	15-3/16
2-5/8	3-7/16	9-1/8	15-1/8
2-3/4	3-5/8	9-1/16	15-1/16
2-7/8	3-3/4	9	15
3	3-15/16	8-15/16	14-15/16
3-1/8	4-1/8	9-15/16	15-15/16
3-1/4	4-1/4	9-7/8	15-7/8
3-3/8	4-7/16	9-13/16	15-13/16
3-1/2	4-5/8	9-3/4	15-3/4
3-5/8	4-3/4	9-11/16	15-11/16
3-3/4	4-15/16	9-5/8	15-5/8
3-7/8	5-1/16	9-9/16	15-9/16
4	5-1/4	9-1/2	15-1/2
		N=½ inch for all Travels	N=6½ inches for all Travels

Light-Duty or Marine Type

Total Travel, Inches	*Load Line "L," Inches*	*"Take-Out" Dimension "C" for Calculating Rod Lengths*	
		Without Bracket, Inches	*With Bracket, Inches*
1-1/4	1-9/16	4-7/16	9-7/16
1-3/8	1-3/4	4-3/8	9-3/8
1-1/2	1-7/8	4-5/16	9-5/16
1-5/8	2-1/16	4-1/4	9-1/4
1-3/4	2-1/4	4-3/16	9-3/16
1-7/8	2-3/8	4-1/8	9-1/8
2	2-1/2	4-1/16	9-1/16
2-1/8	2-11/16	4	9
2-1/4	2-7/8	3-15/16	8-15/16
2-3/8	3	3-7/8	8-7/8
2-1/2	3-3/16	3-13/16	8-13/16
		N=¼ inch for all Travels	N=5¼ inches for all Travels

NOTE: $C_h = C \pm T_a/2$ in which $+ T_a$ is the actual travel "UP" and $- T_a$ is the actual travel "DOWN" from "COLD" to "HOT".

TABLE II—BELLCRANK-AND-SPRING HANGER (FIGURE 8).

SAFE MAXIMUM HANGER ROD LOADS.

Safe Maximum Load, Pounds*	*Diameter Rod in Inches*					
	3 Feet or Less in Length		*Between 3 Feet and 9 Feet Long*		*9 Feet or Greater in Length*	
	J	*K*	*J*	*K*	*J*	*K*
2,500.	¾	½	⅞	¾	1	⅞
3,500.	⅞	¾	1	¾	1⅛	1
4,500.	1	¾	1⅛	⅞	1¼	1
5,500.	1⅛	⅞	1¼	1	1¼	1⅛
7,000.	1¼	1	1½	1⅛	1½	1⅛
10,000.	1½	1⅛	1½	1⅛	1½	1⅛

* Since it is safe to assume that the temperature of a hanger rod is never greater than half the internal pipe temperature, these size rods are safe for an internal pipe temperature of 1000 degrees F. For the longer rods greater diameters are necessary to insure rigidity.

TABLE III—BELLCRANK-AND-SPRING HANGER (FIGURE 8).

LOAD-TRAVEL PRODUCT RANGES—HEAVY DUTY TYPE.

Hanger No.	*Spring Unit No.*	*Load-Travel Product,* Pound-Inches*		
		"Minimum"	*"Normal"*	*"Maximum"*
1	1	988	1,190	1,416
	2	1,347	1,619	1,925
	3	1,912	2,297	2,732
2	4	2,821	3,391	4,030
	5	3,973	4,775	5,680
	6	5,147	6,187	7,358
3	7	6,782	8,151	9,695
	8	8,125	9,766	11,615
	9	9,789	11,766	13,993
4	10	11,644	13,996	16,646
	11	12,736	15,299	18,173

* Total Travel of hanger ranges from 2 to 4 inches in steps of ⅛ inch. To obtain load for a given Total Travel, divide the "LOAD-TRAVEL PRODUCT" by the Total Travel. To obtain the maximum load adjustability in the field, select load from the "NORMAL" column. Always select Total Travel greater than Actual Travel by at least ¼ inch if possible. "MINIMUM" corresponds to 16.88 per cent decrease and "MAXIMUM" to 18.80 per cent increase with respect to the "RATED CAPACITY" at "NORMAL" (zero) on load scale.

Test "LOAD-TRAVEL PRODUCT" figures will vary plus or minus 4 per cent from those given in the Table above, due principally to variations in the load deflection rates of the springs.

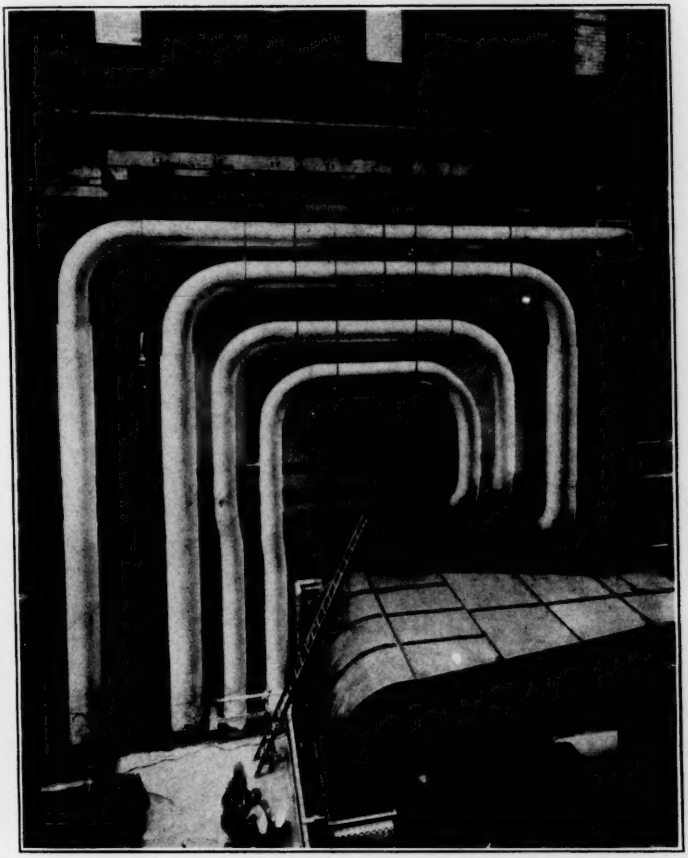

FIGURE 9.—CONSTANT SUPPORT BELLCRANK-AND-SPRING HANGERS SUPPORTING MAIN STEAM LINES IN WATERSIDE STATION OF THE CONSOLIDATED EDISON COMPANY OF NEW YORK, INC.

TABLE IV—BELLCRANK-AND-SPRING HANGER (FIGURE 8).

LOAD-TRAVEL PRODUCT RANGES—LIGHT DUTY OR MARINE TYPE.

Hanger No.	*Spring Unit No.*	*Load-Travel Product,* Pound-Inches*		
		"Minimum"	*"Normal"*	*"Maximum"*
1	1	529	637	754
	2	720	868	1026
	3	1021	1230	1455
2	4	1506	1814	2141
	5	2116	2553	3013

* Total Travel of hanger ranges from 1¼ to 2½ inches in steps of ⅛ inch. To obtain load for a given Total Travel, divide the "LOAD-TRAVEL PRODUCT" by the Total Travel. To obtain the maximum load adjustability in the field, select load from the "NORMAL" column. Always select Total Travel greater than Actual Travel by at least ¼ inch if possible. "MINIMUM" corresponds to 16.00 per cent decrease and "MAXIMUM" to 18.16 per cent increase with respect to the "RATED CAPACITY" at "NORMAL" (zero) on load scale.

Test "LOAD-TRAVEL PRODUCT" figures will vary plus or minus 4 per cent from those given in the Table above, due principally to variations in the load deflection rates of the springs.

Steps 4 to 7, inclusive, may be omitted and considerable time and labor saved if the designer's mind can be satisfied that constant support hangers should be used for pipe lines having vertical shifts in excess of ½ inch. There may not always be full justification for this rule but when the non-resonant characteristic of constant support hangers and the complete absence of augmentative weight transfer stresses are considered along with the great amount of time and labor saved this rule seems to be amply justified.

Figure 9 shows a complete constant support hanger installation in a recent modern central power plant.

CONTROLLABLE PITCH PROPELLER.

By J. H. Strandell.*

The controllable pitch propeller, which has attained great importance in aviation, may have an equally important part to play in marine engineering.

The author in this article reviews the history of the controllable pitch propeller, tracing its beginning from successful water turbine applications. His article is a timely follow-up to the one by Dr. Nordstrom which was translated by the author and reprinted in the February 1940 issue of the Journal.

After describing historical designs and applications, the author covers the field of current practice which is almost entirely limited to the Scandinavian countries. This is followed by a proposed design of the author, which is described in detail with particular regard to its application in the field of submarine propulsion.

There can be but little doubt that we are due for considerable experimentation, in this country, with variable pitch propellers. What their future application may be must be left for the future to disclose.

With a fixed pitch propeller on a ship, only one operating function can be efficiently and completely satisfied. This is usually selected for rated speed of the ship, and full horsepower developed by the engines with the ship's displacement in the loaded condition in calm water and no wind. It becomes apparent that when a ship is operating with a fixed pitch propeller, any condition other than the one that the propeller is designed for will not utilize the engine power to its full extent. Head or tail winds will increase or decrease the resistance of the ship with consequent variation in RPM. of the engine due to the alteration of propeller

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torque. If the propeller is designed for maximum efficiency at full speed of the ship, obviously, the angle of attack at all speeds below full speed will be greater than that for maximum efficiency at these lower speeds. If now the condition is to accelerate the ship from zero speed to full speed, the propeller at no point below full ship speed reaches its theoretical RPM., that is: there is a marked reduction in engine RPM. because this fixed angle of attack is greater than the most efficient at all speeds of advance below full speed of the ship. In other words, in all of the above mentioned cases there are losses of power, which means that the available engine power cannot be utilized in many instances of operation when it is connected to a fixed pitch propeller.

As the conditions under which an airplane is working with a fixed pitch propeller are in many respects very similar to the conditions under which a ship is functioning, an airplane provided with a fixed pitch propeller has only one flight function that can be efficiently satisfied. This may be selected as maximum speed with the engine under wide-open throttle condition. Then, all other conditions such as take-off, climb, ceiling, and cruising speed at altitude, and low altitude performance cannot be satisfied with the fixed pitch propeller. The aeronautical engineers found it an absolute necessity to develop the controllable pitch propeller because of the need for better airplane performance. The improvement has been shown to be so great as to well pay for the added complication and expense involved.

The speed of ships has greatly increased in the last few decades and as a consequence large expensive power plants must be installed. Obviously, it is of paramount importance that the power plants be fully utilized for better ship's performance and it is believed that one of the greatest contributions to that effect is the installation of a controllable pitch propeller. Possibilities of improvement in ship's performance in connection with a controllable pitch propeller are even more promising when it is realized that the reversing element of the power plant is not needed, thus making a greatly simplified machinery plant and reducing the time of reversing or changing the speed of the ship.

On the basis of past experience with small craft using controllable pitch propellers, and the already successful installation of

improved designs of controllable pitch propellers in large foreign commercial ships as well as naval vessels, it is anticipated that further development and research in this line will prove itself to be just as important as it has been for airplanes.

The first usable controllable pitch propeller was designed in the middle of the nineteenth century by an Englishman named Bevis, Figure 1. The characteristic feature of this design is that the rod for changing the pitch is located in the drilled-out propeller shaft. The rod regulates the pitch by means of a movable collar fitted on the shaft. The mechanism for small propellers is moved by hand and for larger ones by hydraulic servomotor. For changing the pitch, large forces are required, even greater than the propeller thrust itself. These forces have to be taken up by the collar as well as by the thrust bearing for the shaft, which is a great disadvantage.

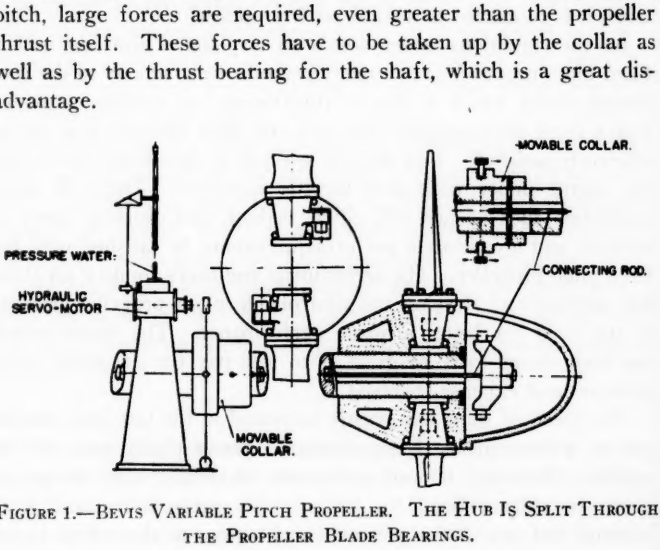

FIGURE 1.—BEVIS VARIABLE PITCH PROPELLER. THE HUB IS SPLIT THROUGH THE PROPELLER BLADE BEARINGS.

A number of different designs have lately been developed, all more or less based on Bevis' design. The difficulties have always been to transmit the large forces involved to the rotating collar. In order to decrease the risk of overheating the collar it is provided with ball bearings, and in some instances a mechanical servomotor rotating with the shaft is provided. These forces, of course, are in addition to the usual thrust forces on the blades. The servomotors are heavy, occupy a large space and are subject

to rapid wear, thus seriously limiting their extensive use. Even the hub design causes a great deal of trouble. In order to reduce manufacturing costs, the hubs are split through the bearings of the blades so that it is practically impossible to keep them water tight. The manufacturers recommend that the hubs be filled with grease for lubrication of the bearings, but the first time the blades are moved, after installation, grease is squeezed out through the split joint and through the bearings. When the blades are next moved in the opposite direction, water is sucked into the newly created space. This process continues until the hub is filled with a mixture of water, silt, and grease. Obviously, the lubrication quickly becomes ineffective and the bearings wear loose in a relatively short time. The consequence is that the blades vibrate in certain positions where they are supposed to be balanced, adding further to the wear.

In spite of these disadvantages and imperfections the controllable pitch propeller has come to a very extensive use in fishing craft and sailing vessels with horsepower up to 150 and 200. It has been necessary to use controllable pitch propellers in motor driven sailing vessels and trawlers because of the low speed at which the ship must be kept when fishing. Controllable pitch propellers in motor driven sailing vessels have been preferred, partly because the pitch can be increased during sailing and light load condition, in order to better utilize the motor, and partly because the controllable pitch propeller is considered more dependable for reversing than the engine. The "skipper" used to say that the coupling always slipped when it was most needed. The friction coupling which is connecting the reversible engine to the propeller shaft must now and then be adjusted for wear. This is quite often overlooked. (The conventional friction coupling used in these small power plants is very much exposed to slipping, especially when subjected to the sudden shocks introduced during reversing an engine connected to a fixed pitch propeller.)

The question may arise as to why the controllable pitch propeller has not until very recently been extended to use in large ships. Among the reasons are: first, the imperfections of those early designs mentioned above and the tendency of men devoting themselves to a highly specialized art to become reluctant to try

anything which deviates from the conventional, and generally approved, design. Second: no real research in this subject has been made until the last ten or twelve years. And third: the great advance of the controllable pitch propeller in the last fifteen years applied to airplanes has only now started to direct the minds of Naval architects and propeller experts to the idea that it might have similar advantages for ships.

One of the latest investigations in this subject is Doctor Nordstrom's article, "Controllable Pitch Propellers," reprinted in the Notes of the February 1940 issue of the JOURNAL of the AMERICAN SOCIETY OF NAVAL ENGINEERS, in a translation by the writer from the Swedish technical paper, "Teknisk Tidskrift," July 16, 1938, issue 28. Doctor Nordstrom's investigation is a very exhaustive analysis of what can be gained in speed and thrust of a ship when using a controllable pitch propeller. It is believed that his investigation in this subject is one of the most thorough ever made so far published and we are grateful to him for his work. In this article the writer will use some of his charts and refer now and then to his findings. The reader interested in this subject will do well to read the reference cited above to familiarize himself with this investigation and the principle of the charts.

So far as is known the only successful controllable pitch propeller in use today in large ships is that manufactured by Aktiebolaget Karlstads Mekaniska Werkstad, Karlstad, Sweden, and used on several Scandinavian seagoing Merchant and Government ships. The very first installation, that of a motor schooner, will be described here because the ship has been in operation for over two and a half years and its performance has far exceeded the expectations of military and civilian experts. The Swedish design is entirely based on the Kaplan turbine design which has a history of many years of excellent performance under very severe working conditions. Therefore, it is first necessary to take brief notice of the development and design of the Kaplan turbine as developed in Sweden. I prefer to translate and quote directly Mr. Englesson's* description of the Kaplan turbine design from his article the "Ka Me Wa—Propellern" in Teknisk Tidskrift, 1938, numbers 28, 33 and 37. Naval Architecture 7, 8 and 9.

* Mr. Elov Englesson is Director of Engineering at Karlstad Mekaniska Werkstad, Sweden, and is internationally distinguished as a hydraulic turbine expert.

The Kaplan Turbine.

"In 1913, Doctor Viktor Kaplan applied for a patent on a water turbine provided with radially kept blades without an exterior ring and arranged in such a way that the position of the blades could be varied during the operation of the turbine. In other words, it was a ship propeller, with controllable pitch, used as a turbine. Kaplan's invention was regarded by the turbine manufacturers as too daring and was considered impossible to make. They recognized its theoretical correctness, but considered it impossible to design a hub strong enough and dependable enough for such a wheel. The blades which would be subjected to tremendous forces from the water as well as great centrifugal forces could be supported by means of a bearing in only one end. In addition, the position of the blades would be continuously changed in relation to the load of the generator.

"Professor Kaplan was able to find a firm in Brün, named Ignaz Storck, which had never before manufactured turbines, to undertake the manufacture of his water turbine.

"A-B Karlstads Mekantska Werkstad started the manufacturing of Kaplan-turbines in 1921 in their factory in Kristinehamn, Sweden. By that time Storck had made several small units, some with good results and others with less success. KMW's first order was an 11,200 HP. turbine for the Swedish Government to be placed at Lilla Edet in Gota river, Figure 2. This turbine was not only larger than any turbine, of the Kaplan type, made before, but also the largest one of any type. The wheel was to be 6 meters in diameter and have a weight of 62 tons. This order was the result of a very daring decision, but the Swedish Flood Control Commission did the Swedish turbine industry, and at the same time themselves, a great service. The prophesies of our European Technical Colleges were not any too cheerful. When Professor Kaplan, as a result of this order, recommended that others start building Kaplan turbines, the answer was: 'The Kaplan turbines is correct theoretically but from a practical point of view impossible to make.' The answer of the Kaplan representative was: 'But the Swedes are making the biggest turbines in the world as Kaplan turbines.' 'But wait until you see what Lilla Edet will reveal,' was the rejoinder of the representative for the turbine companies.

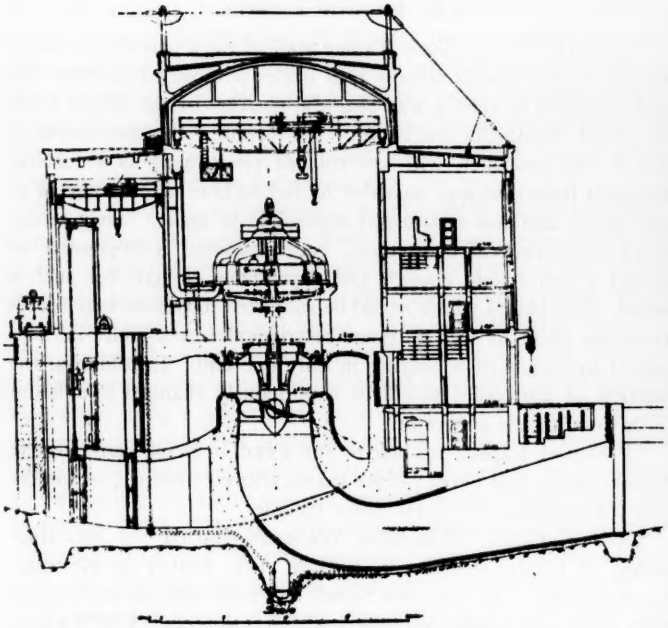

FIGURE 2.—SECTION THROUGH LILLA EDETS' POWER STATION.

"The building of the 'Lilla Edet' power plant was very much delayed because of labor trouble which gave us plenty of time for experiment. In order to obtain a real trial of the hub design a testing rig was built, Figure 3, consisting of a testing wheel where the hydraulic and centrifugal forces were replaced by corresponding spring forces. In this testing rig the blades were moving day and night for many months. Suitable bearing metals and the maximum bearing pressures were determined. In order to investigate the cavitation safety of the wheels a special testing room was built for studying this phenomenon about which at that time very little was known.

"The starting of the 'Lilla Edet' 'Kaplan' turbines which had been looked forward to by hydraulic engineers all over the world, took place at the end of the year 1925, and in the summer of 1926 a commission of experts made a final trial of the power plant.

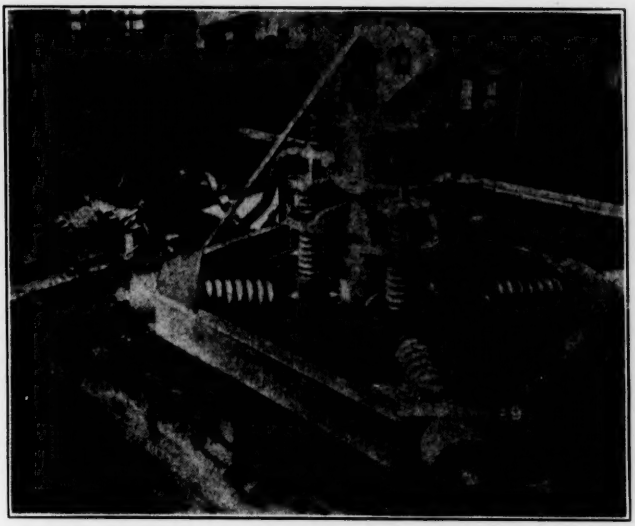

FIGURE 3.—TEST RIG FOR A KAPLAN HUB. THE RADIALLY APPLIED SPRINGS REPRESENT THE CENTRIFUGAL FORCE, AND THE VERTICAL SPRINGS THE WATER PRESSURE, ON THE BLADES. SUITABLE BEARING METAL AND BEARING PRESSURES WERE TESTED IN THIS RIG.

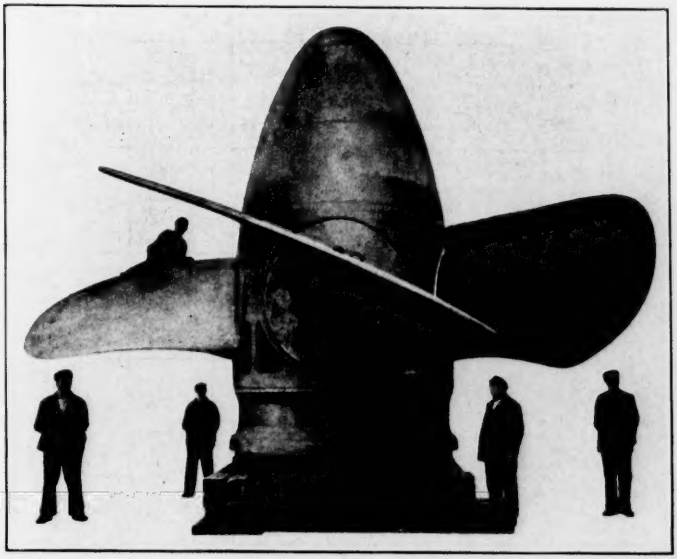

FIGURE 5.—8-METER DIAMETER TURBINE WHEEL FOR VARGONS POWER STATION. THE LARGEST TURBINE WHEEL IN THE WORLD. THE WEIGHT OF EACH BLADE IS 10.5 TONS.

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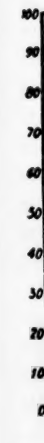

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The results were excellent, verifying completely the test data obtained at the factory. See Figure 4. The turbine could be overloaded to 14,000 HP. instead of the guaranteed 11,200 and a maximum efficiency of 92.5 per cent was reached. Moreover, one could not ask for better dependability. As soon as the test results were known, the Kaplan turbine became very popular. Numerous firms in Sweden as well as all over the world bought the license for the manufacture of this type, which, at present, is completely predominant within the range of water fall for which it is suited. I will mention that at K. M. W. during the last four years the total number of horsepower of Kaplan turbines sold and on order has been double that of all the other types.

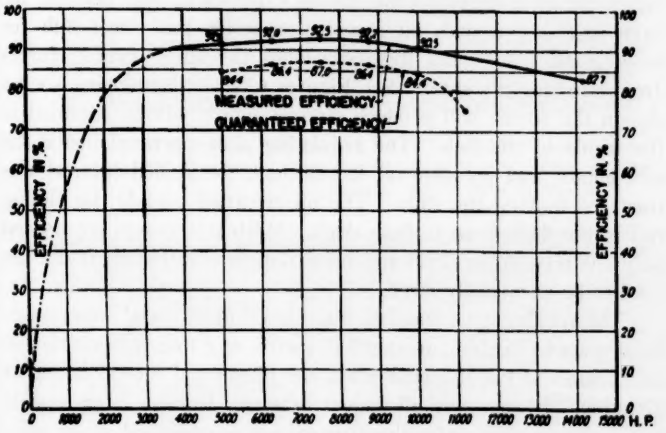


FIGURE 4.—EFFICIENCY CURVE OBTAINED DURING TEST AT "LILLA EDET" BEFORE DELIVERY TO THE SWEDISH GOVERNMENT.

"The Lilla Edet turbine size has been surpassed many times by both K. M. W. and other firms. The largest Kaplan turbine wheels in the world were those made for the Swedish government station at 'Vargon.' The turbine wheel is shown in Figure 5 and has a diameter of 8 meters. The next largest turbines are the four of 37,500 HP. each which have been delivered to the Swir power

station in Russia. These turbine wheels are 7.42 meters in diameter. There are under construction four turbines of close to 50,000 HP. each.

AUTHOR'S NOTE: After this was published an order was placed by the Swedish government for a Kaplan turbine of 63,000 HP.

K. M. W.'s HUB DESIGN FOR KAPLAN TURBINE.

"Thanks to the necessity of building in the very beginning a turbine of large size, we had at the very outset to solve the problem of a hub design to overcome tremendous forces when turning the blades. Consequently it was not necessary for us to go through all the many stages of development in the hub design which the other turbine firms had to experience. We laid out the very first design with oil servomotor inside the wheel hub for moving the blades and thus prevented the large turning forces from stressing the shaft. See Figures 6, 7, 8. Employing such a design the forces will produce stresses which will be confined to the inside of the hub. The regulating slide operated inside the servomotor and pressure oil fed through the drilled-out rod was used for moving the slide. The oil returned outside the sliding rod in the drilled-out turbine shaft. With this design we needed only one pipe in the shaft and the drilled hole in the shaft did not need to be excessively large.

"The regulating mechanism was placed in a special water insulated space in the hub, arranged in such a way that the pressure in that space did not decrease when the piston rod moved in either direction. In that way all danger of water leakage from outside was removed. We have, therefore, never experienced water mixing with the oil in the hub. Thanks to long and fastidious testing of bearing metals and determination of permissible bearing pressures in the hub testing rigging, we have been able to keep down the wear. A couple of years ago a check-up was made on the bearing clearances at power plants which have been in operation for more than ten years, and it was found that the bearing clearances in all cases were under 1/10 mm. At the 'Lilla Edet' power plant, the measured clearances were 0.04 mm., practically the same as the original clearances.

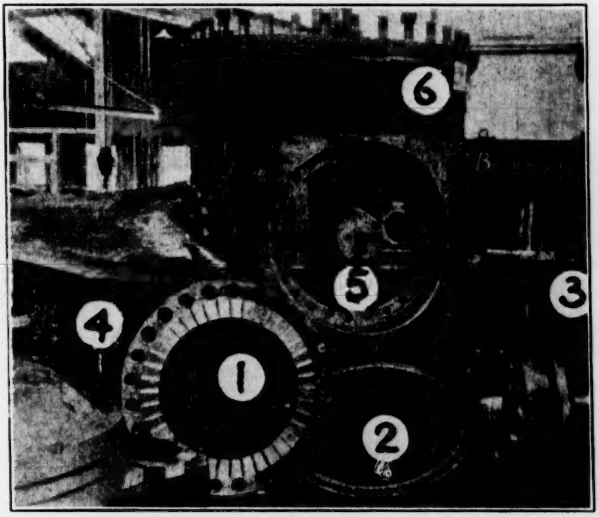


FIGURE 7.—THE HUB OF A 34000 HP. "KAPLAN" TURBINE USED AT LANGHAGS' POWER STATION. ① BEARING RING, ② SEALING RING, ③ CRANK-PIN RING, ④ PISTON, ⑤ CENTER PIN, AND ⑥ SLIDING BLOCK.

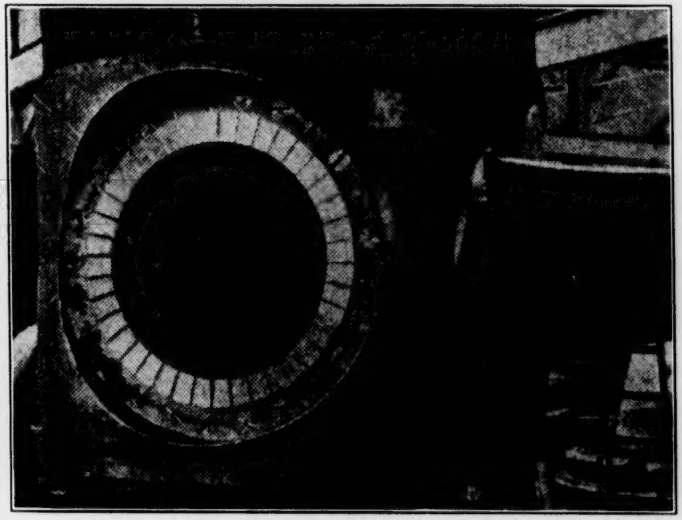


FIGURE 8.—THE SAME HUB, SHOWN IN FIGURE 9, WITH CRANK PIN AND BEARING RING ASSEMBLED IN PLACE.

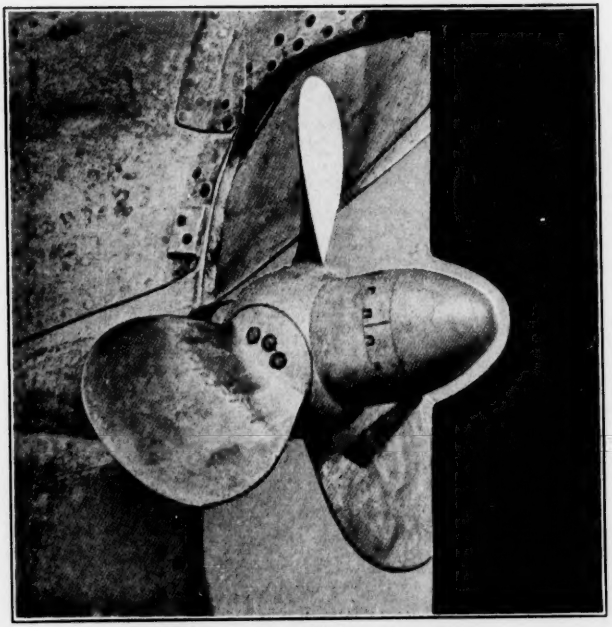


FIGURE 9.—KA MÉ WA PROPELLER.

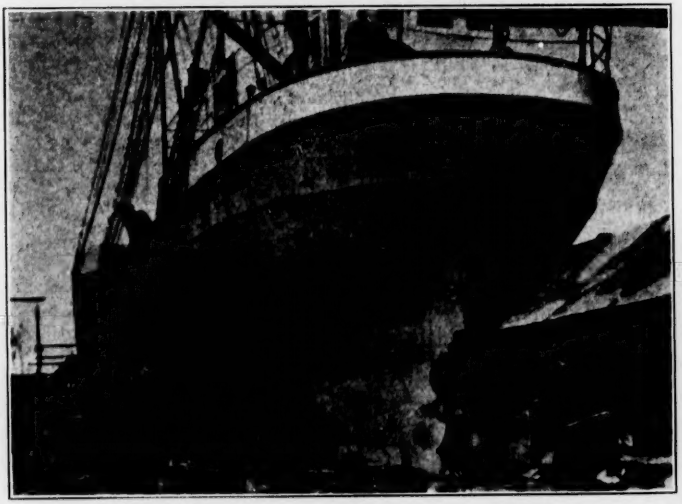


FIGURE 10.—THE MOTOR-SCHOONER "RANE," WEIGHT 100 TONS, FREIGHT 285 TON, 150 HP. DIESEL ENGINE. SPEED OF THE BOAT EQUIPPED WITH KA MÉ WA PROPELLER, 9.85 KNOTS.

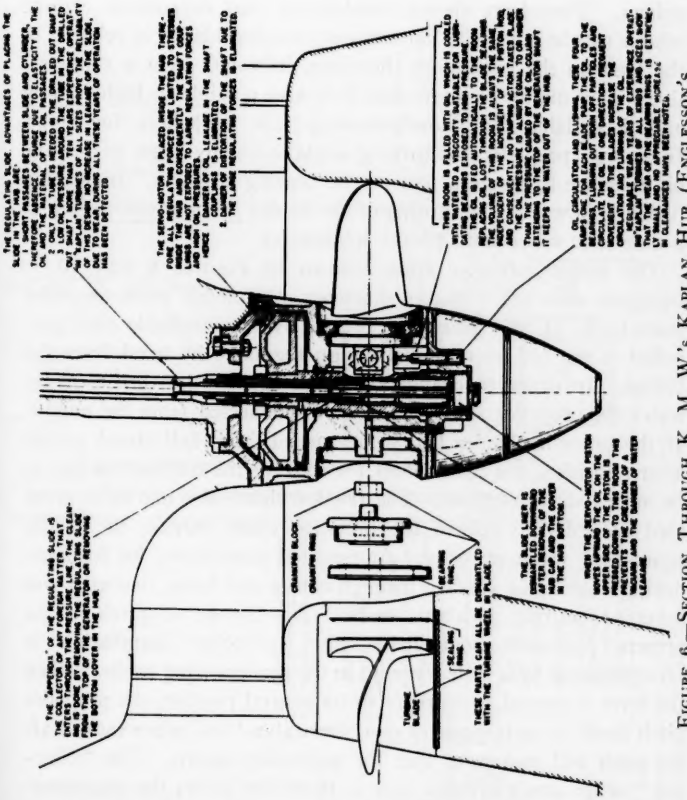


FIGURE 6.—SECTION THROUGH K. M. W.'s KAPLAN HUB (ENGLESSON'S PATENT).

"Thanks to favorable conditions and a well directed procedure of designing and testing, our hub design was from the beginning thoroughly worked out to fit all sizes of turbine wheels. During all these fifteen years no essential changes have been required."

THE EXPERIENCE OF THE KAPLAN TURBINES APPLIED TO THE PROPELLER OF A SHIP.

It is quite natural that a concern like the Swedish hydraulic turbine firm with its many years of experience with Kaplan turbines should take up manufacturing of controllable pitch pro-

pellers. They have shown conclusively that dependable turbine wheels can be built with continuously moving blades in relation to the load on the generator; therefore, it would seem a comparatively easy matter to show that it is also possible to build strong and dependable continuously-moving-blade propellers for ships. In a ship's propeller, the turning angle is usually large, giving the oil a chance to create a film on the bearing surfaces. In Kaplan turbines, however, the turning of the blades is about one degree at a time with consequent poorer lubrication.

The motor schooner *Rane,* shown in Figures 9 and 10, is equipped with a 1.5 meters diameter controllable pitch propeller made by K. M. W. *Rane* had, previously, a controllable pitch propeller of the old conventional design regulated by hand from the bridge. In order to change the pitch against the thrust of the water the propeller shaft had to be disconnected from the engine. If the propeller blades had to be turned from full ahead to full astern position, the hand-wheel for moving the mechanism had to be turned six revolutions. The friction clutch also had to be operated in order to relieve the pressure when moving the blades against the water pressure. At repeated maneuvers, for instance, during docking or passage through canals and locks, one man was occupied entirely with this work. The change of pitch in the present hydraulic controllable pitch propeller installation is brought about by a little lever up in the maneuvering cabin. When the lever is moved forward from its neutral position the propeller pitch is set to correspond to movement ahead and when moved aft the pitch will change to that for movement astern. The " Skipper " never needs to think how to move the lever; the maneuvering will be, with a little exercise, automatic. In small vessels of the *Rane's* size the power plant takes care of itself. The engine's speed is regulated by a governor which is adjusted from the control cabin to vary the engine speed as required. The controllable pitch propeller as well as the regulating mechanism is automatically lubricated, so that no periodical oiling is necessary.

The Hub Design of Mr. Englesson's Propeller.

Figure 11 shows the installation of the Englesson propeller in the motorschooner *Rane.* The propeller hub is shown in cross section indicating the principle of the working mechanism. The

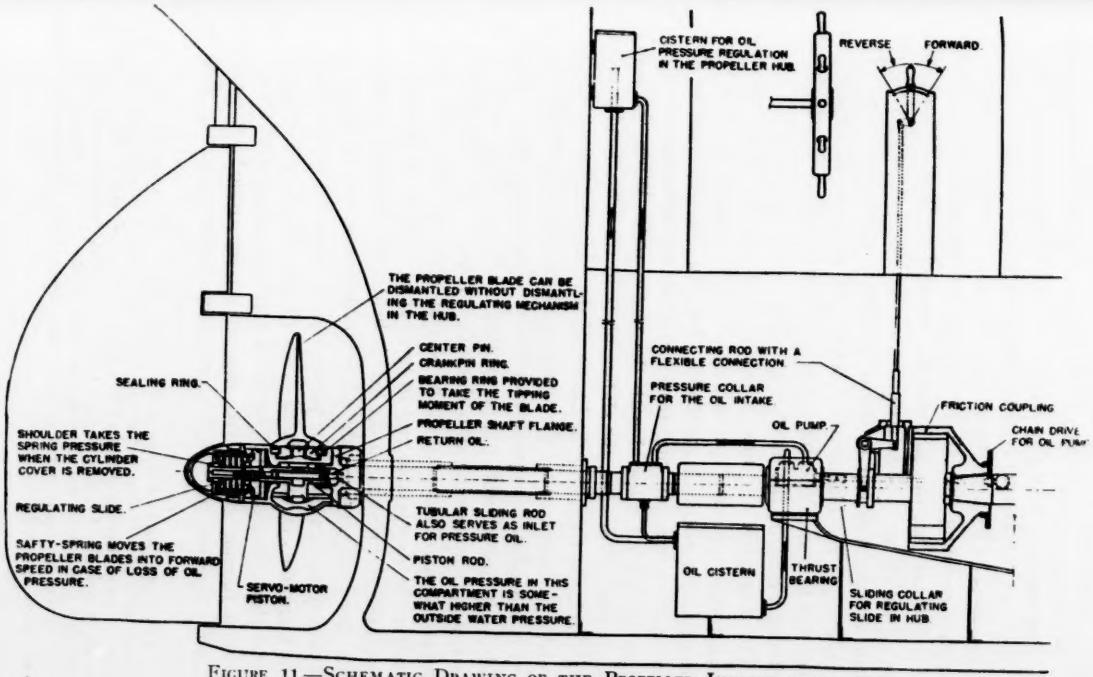

CISTERN FOR OIL PRESSURE REGULATION IN THE PROPELLER HUB.
REVERSE
FORWARD
THE PROPELLER BLADE CAN BE DISMANTLED WITHOUT DISMANTLING THE REGULATING MECHANISM IN THE HUB.
CENTER PIN.
CRANKPIN RING.
BEARING RING PROVIDED TO TAKE THE TIPPING MOMENT OF THE BLADE.
PROPELLER SHAFT FLANGE.
RETURN OIL.
SEALING RING.
SHOULDER TAKES THE SPRING PRESSURE WHEN THE CYLINDER COVER IS REMOVED.
REGULATING SLIDE.
SAFTY-SPRING MOVES THE PROPELLER BLADES INTO FORWARD SPEED IN CASE OF LOSS OF OIL PRESSURE.
SERVO-MOTOR PISTON.
TUBULAR SLIDING ROD ALSO SERVES AS INLET FOR PRESSURE OIL.
PISTON ROD.
THE OIL PRESSURE IN THIS COMPARTMENT IS SOMEWHAT HIGHER THAN THE OUTSIDE WATER PRESSURE.
CONNECTING ROD WITH A FLEXIBLE CONNECTION.
PRESSURE COLLAR FOR THE OIL INTAKE.
OIL PUMP.
FRICTION COUPLING.
CHAIN DRIVE FOR OIL PUMP.
OIL CISTERN
THRUST BEARING
SLIDING COLLAR FOR REGULATING SLIDE IN HUB.

FIGURE 11.—SCHEMATIC DRAWING OF THE PROPELLER INSTALLATION IN A SAILING VESSEL.

hub of the propeller is, in principle, of the same design as the hub of the Kaplan turbine. The blades are held to the so-called crank-pin ring with six large cap screws. If it is necessary to change a blade or to inspect the bearing ring and oil seal ring, all that is necessary is to remove the six cap screws. There is no necessity for disassembling the hub or the regulating mechanism. Under the condition that the ship can be trimmed to show at least half the propeller hub above water, it is obvious that inspection and blade replacement can be made without docking the ship. The anchoring of the blades in the hub is of such sturdy dimensions that should any blade hit a hard inflexible object it will break without damaging the hub.*

The crank pin ring moves around a center pin which takes the axial forces, and also against an exterior bearing ring which takes the tipping moment of the blade. See also Figures 7 and 8. The pin on the crank pin ring works in a sliding block moving in the cross head built integral with the piston rod. When the piston rod is moved forward, the sliding block in the cross head revolves the crank pin ring, and the blades of the propeller are set for forward movement. Just the opposite takes place when the piston rod is moved in the opposite direction. Each blade flange is provided with an oil seal ring preventing oil from leaking out as well as water from coming into the hub. This oil seal ring is of the same design as that used in the Kaplan turbines, and it has shown itself to be dependable and to have good sealing properties. Because the oil pressure in the hub is held somewhat higher than the outside water pressure, the oil is enabled to keep bearing surfaces copiously lubricated. The piston rod is connected to the piston in the hydraulic cylinder placed in the propeller hub. As in the Kaplan turbine the piston rod carries a regulating slide which is moved with a tubular sliding rod inserted in the drilled-out propeller shaft. The pressure oil is transmitted to the regulating slide by

* Such a case happened in one of the Swedish Kaplan turbine installations. Due to circumstances too involved to mention here, there occurred during a sudden change from no load to full load of 16,000 HP. a water hammer blow, that is: the water column in the suction pipe was torn apart and then forced back by the atmospheric pressure with such terrific force that all of the four blades broke off at the flanges. However, the hub was not damaged and the only repair required was to unscrew the flanges and replace them with new blades.

means of the tubular sliding rod. The regulating slide is arranged in such a way that when the slide is moved forward the oil is admitted to the after side of the piston and the forward side is then connected to the outlet outside of the sliding tube in the drilled-out propeller shaft. The oil pressure moves the piston in the same direction and through the same distance as the regulating slide moves. If the regulating slide is moved aft, the pressure oil is admitted to the forward side of the piston and the other side is now connected to the outlet causing an aft movement of the piston. This can be described as a piston hydraulically connected to the regulating slide and its sliding rod. In comparison with the Kaplan turbine this is a slight improvement on the regulating slide for the ship's propeller. For any given movement of the maneuvering lever on the bridge, the stroke of the piston rod which is the same as that of the regulating slide has been limited to a very short distance in either direction. If an effort is made to move the regulating slide further it will not move faster than the piston whose motion follows the slide's motion exactly. As the propeller blades follow the movement of the regulating lever on the bridge there is always an indication of the piston position and consequent position of the blades shown by this regulating lever.

In this connection it should also be pointed out that the regulating slide design in the hub, always used in the Englesson Kaplan turbine design, has not only the advantage of using one tube instead of two in the drilled-out shaft, but also gives the oil a chance to cool when pumped through the long shaft and the hub. No special cooler for the oil is needed. The regulating slide is made with a little negative overlap so that when it is in the middle position, the oil will leak by. Another difference is the spring pressing on the piston. If the oil pressure should fail, this spring will move the piston to the position of forward speed if the piston is not already in that position, in which case it will be kept there. However, the spring is not sufficiently strong to reverse the pitch while under the engine load. Therefore, the engine must be stopped before the spring can act. The spring arrangement is designed in such a way that when the blades are in forward position the after spring holder rests on a shoulder integral with the

piston rod. The hub cap can then be removed without dismantling the spring. A collar, which does not rotate with the shaft, is located close to the stern tube, and acts as a distributor taking the pressure oil into the shaft and the propeller hub by way of the tubular sliding rod, and, also, leading the oil out from the hub and the drilled-out shaft. The collar design has been very thoroughly tested in the manufacturer's shop and has shown itself to be satisfactory in operation on the ship. The return oil from the propeller hub is piped to an oil tank located over the water line in order to give an oil pressure in the hub somewhat higher than the surrounding water pressure. A pump, driven from the engine, supplies the necessary pressure for maneuvering. On larger ships a directly driven pump with sufficient capacity for slow and small changes in pitch of the propeller is satisfactory. A larger electrically driven oil pump is automatically started when actual maneuvering is required.

Figure 12 shows a proposed installation of Mr. Englesson's propeller in a 16,000-ton oil tanker. Since Mr. Englesson published this proposal two years ago, he has installed his propeller in several large seagoing ships and the experience to date with these installations is highly satisfactory.

Just before this article was sent for printing, the writer received a letter from Mr. Englesson enclosing a catalogue and advertising material for his controllable pitch propeller. He also forwarded an installation drawing with notes printed in English. The installation drawing, three of the pictures from the catalogue, and a free translation of the advertising are shown on the following pages.

The design of the propeller appears to be in principle the same as that installed in the motor schooner *Rane*, and described previously, see Figure 11, with the exception that it is further developed to suit any size ship and power plant. Figure 13 shows a typical installation now being installed in a large number of ships such as tow-boats, survey ships, tankers, freighters, mine sweepers, etc., several of which are already in operation. In Figure 14 is shown a photograph of the assembled propeller, the propeller shaft, the casing over the operating valve rod, the distributing box for supply and return oil and the auxiliary servomotor for operat-

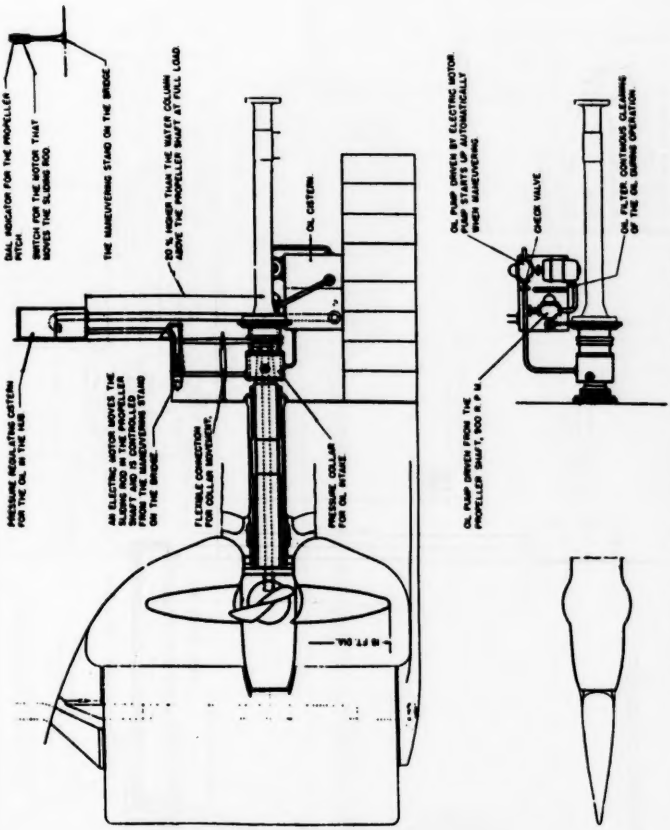

DIAL INDICATOR FOR THE PROPELLER PITCH
SWITCH FOR THE MOTOR THAT MOVES THE SLIDING ROD
THE MANEUVERING STAND ON THE BRIDGE
20 % HIGHER THAN THE WATER COLUMN ABOVE THE PROPELLER SHAFT AT FULL LOAD
OIL CISTERN
OIL PUMP DRIVEN BY ELECTRIC MOTOR. PUMP STARTS UP AUTOMATICALLY WHEN MANEUVERING
CHECK VALVE
OIL FILTER CONTINUOUS CLEANING OF THE OIL DURING OPERATION.
PRESSURE REGULATING CISTERN FOR THE OIL IN THE HUB
AN ELECTRIC MOTOR MOVES THE SLIDING ROD IN THE PROPELLER SHAFT AND IS CONTROLLED FROM THE MANEUVERING STAND ON THE BRIDGE
FLEXIBLE CONNECTION FOR COLLAR MOVEMENT.
PRESSURE COLLAR FOR OIL INTAKE
OIL PUMP DRIVEN FROM THE PROPELLER SHAFT, 900 R. P. M.

FIGURE 12.—PROPOSED INSTALLATION OF A KA MÉ WA PROPELLER IN A 16,000-TON OIL TANKER.

ing the valve rod in the shaft. In Figure 15 is shown a photograph of the propeller with the following parts removed and shown on the table in the left side of the pitcture: ① Propeller blade, ② Sealing ring, ③ Crank pin ring, ④ Bearing ring, ⑤ Crank pin sliding block, ⑥ Safety springs. The first five parts can be removed from the propeller without further dismantling of the hub. In Figure 16 is shown a photograph of the propeller with the cone piece removed. Parts of the safety springs in the oil

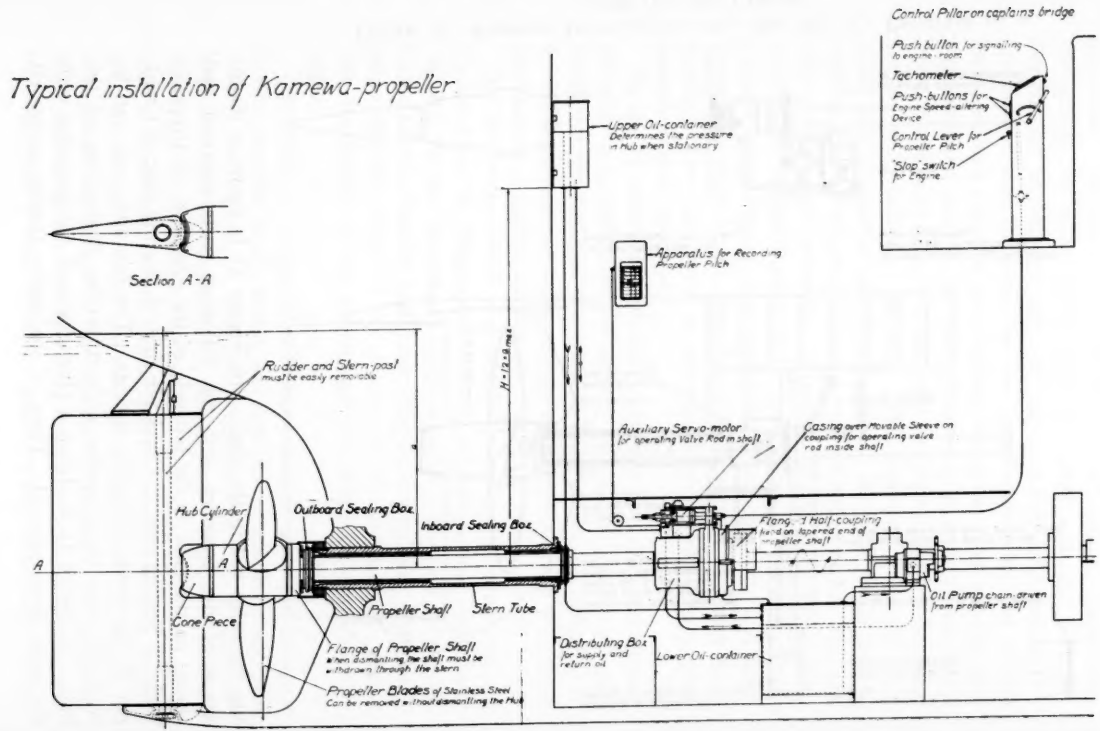

Typical installation of Kamewa-propeller
Control Pillar on captains bridge
Push button for signalling to engine room
Tachometer
Push-buttons for Engine Speed-altering Device
Control Lever for Propeller Pitch
'Stop' switch for Engine
Upper Oil-container Determines the pressure in Hub when stationary
Apparatus for Recording Propeller Pitch
Section A-A
Rudder and Stern-post must be easily removable
H-12-9 max
Auxiliary Servo-motor for operating Valve Rod in shaft
Casing over Movable Sleeve on coupling for operating valve rod inside shaft
Hub Cylinder
Outboard Sealing Box
Inboard Sealing Box
Flang. 'd Half-coupling keyed on tapered end of propeller shaft
A
A
Oil Pump chain-driven from propeller shaft
Cone Piece
Propeller Shaft
Stern Tube
Flange of Propeller Shaft When dismantling the shaft must be withdrawn through the stern
Distributing Box for supply and return oil
Lower Oil-container
Propeller Blades of Stainless Steel Can be removed without dismantling the Hub

FIGURE 13.

FIGURE 13.

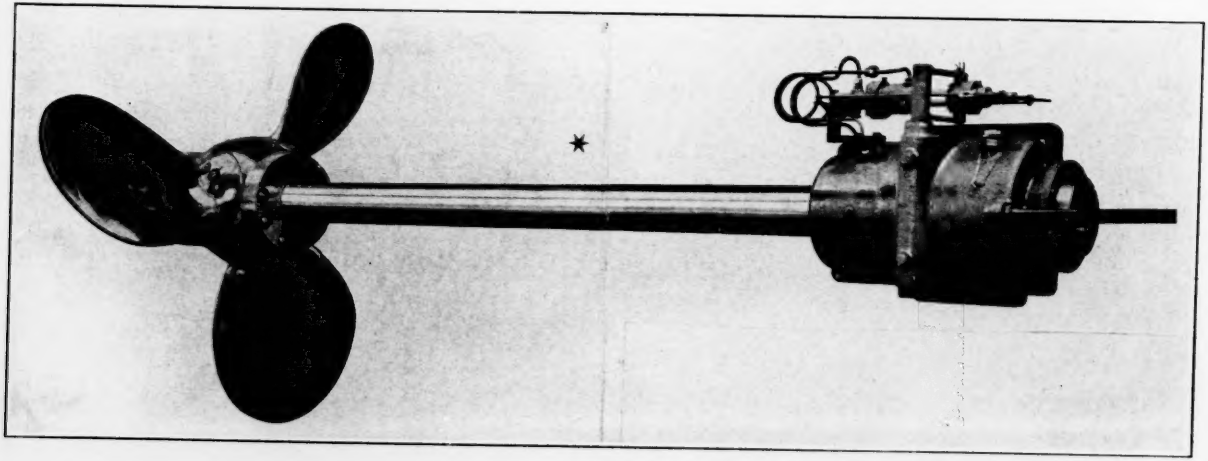


FIGURE 14.

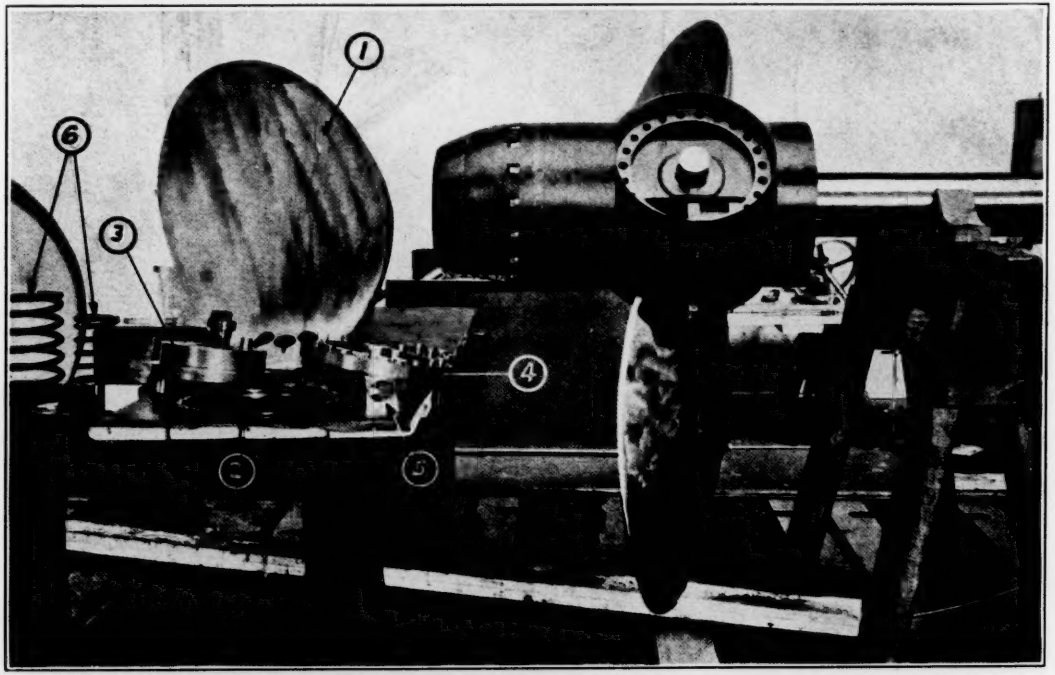


FIGURE 15.

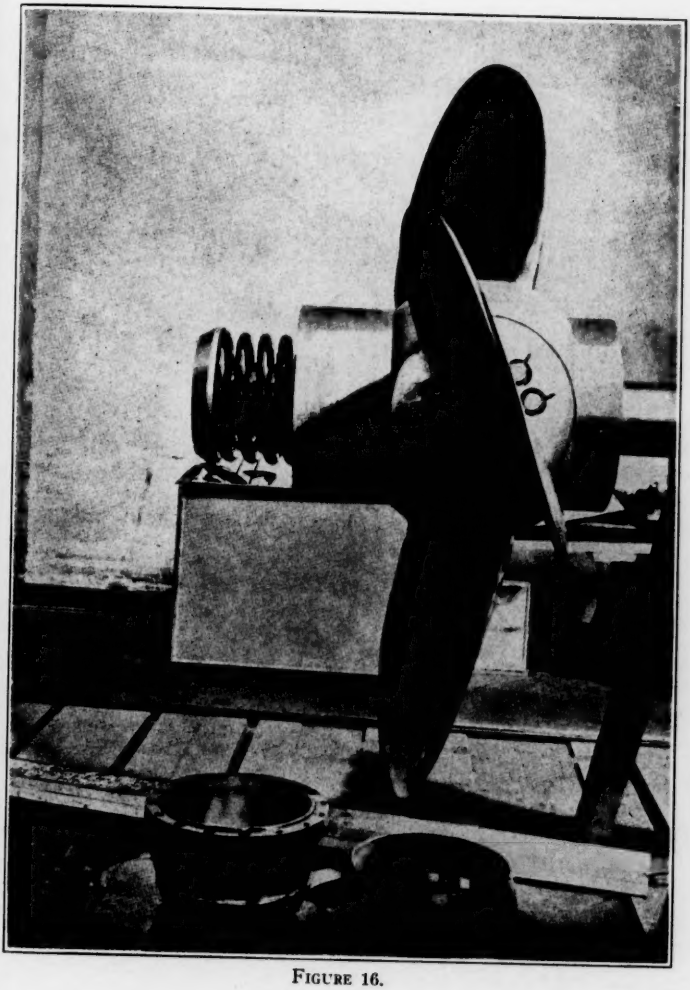
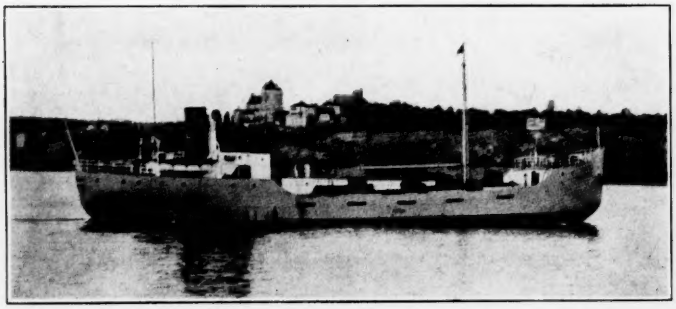


FIGURE 16.

The Ka Mé Wa Propellern Makes the Diesel Engine an Ideal Power-Plant for all Kinds of Ships Because



M/S Dalanas.

The Power-Plant: 400 HP. Diesel Engine Connected to a Ka Mé Wa Propeller.

① Any desired speed of the ship from maximum to zero may be obtained.
② The maximum horsepower of the Diesel engine may, under all conditions, be utilized.
③ All maneuvering takes place from the bridge, thus, reducing the possibility of maneuvering errors.
④ Decreased wear of the cylinders and less risk of cracking cylinder heads, due to elimination of reversing and the greatly reduced number of air starts; results in more reliable engine performance.
⑤ Critical speeds of torsional vibration may easily be avoided and a quiet vibration free operation obtained.

The Ka Mé Wa propeller can be manufactured for any size of engine used in ships today.

The design of the Ka Mé Wa propeller is based on our seventeen years' experience with Kaplan turbines. Our firm leads the world in manufacturing this type of turbine.

Figure 17.

No Man in the Engine Room When Danger Exists from Mine Explosions or Torpedoes.

This is a Desire Which, During This Time, Can Be Fulfilled if the Ship Is Equipped with a Ka Mé Wa Propeller, Because All Maneuvering Is Done from the Bridge Without Cooperation of an Engineer.

Note Other Advantages

① The maneuvering ability is superior to any system using fixed propeller blades. The speed of the ship may be regulated to any value from zero to maximum.

② Carbonization of the combustion space, wear of the cylinders, and the risk of cracking the cylinder heads, are reduced because reversing is eliminated and the number of air starts is kept to a minimum.

③ The maximum horsepower of the Diesel engine may, under all conditions, be utilized.

④ Greater possibilities for using excess power of the engine by increasing the RPM.

⑤ The blades of the Ka Mé Wa propeller can be turned with the engine under full power and RPM. Therefore, only one lever is needed to control the ship's speed.

The Ka Mé Wa Propeller Can Be Manufactured for Any Size of Engine Used in Ships Today.

The Design of the Ka Mé Wa Propeller Is Based on Our Many Years of Experience with Kaplan Turbines.

Figure 18.

pressure cylinder are shown. The function of these springs has been described previously in this article.

Mr. Englesson writes that during this exceptionally hard winter his propellers installed in several ships which have been in continuous operation for months traveling in ice have been subjected to severe tests. On one of the propellers the forces were so great that the propeller shaft was bent, and on a tow-boat the propeller hub tore loose from the propeller shaft flange. In both cases no damage to the propeller blades or their bearings occurred. Mr. Englesson mentioned that he has at present 16 propellers under manufacture to be installed in various ships.

In the advertisement reproduced in Figure 17 is a picture of a small freighter commissioned early last spring. Several freighters of this type, equipped with controllable pitch propellers, are in operation in Scandinavian waters today. This very fact should demonstrate to marine engineers that such a device is important not only for ships connected with towing work, as for instance tow-boats, trawlers, mine sweepers and other ships where the maximum speed is far above the cruising speed, but also for other types of vessels. A Scandinavian ship owner must turn over his pennies several times to investigate whether the ends will meet, or if a little profit is possible, before he decides to take a cargo. It is quite certain that he would not employ a device for the purpose of improving the performance of a ship if it did not benefit his business.

The advertisement shown in Figure 18 is quite significant in view of the war psychology developed in Europe today. A ship operating in those waters at present may hardly leave the harbor without exposure to the risk of being sunk. It is, therefore, desirable to have the operating personnel stationed in the safest possible spot in the ship. When we once start manufacturing controllable pitch propellers for the merchant marine, it is sincerely hoped that the American manufacturers will never need to be given an argument such as this for advertising purposes.

The engines shown in Figure 19 are representative types of Bolinder's Company, Inc., low pressure heavy duty marine oil engines. These types of oil engines are designed for driving two-blade reversible propellers. The engines are built in one and two-

① Maneuvering column.
② Hand wheel for turning the propeller blades.
③ Friction clutch.
④ Friction clutch disengaging and engaging lever.
⑤ Propeller with reversible blades.
⑥ Stern tube.
⑦ Hit and miss governor.
⑧ Centrifugal governor.
⑨ Propeller blades integral with turning cranks.
⑩ Cross head for moving propeller blade cranks.
⑪ Crank pins.
⑫ Propeller shaft.
⑬ Sliding sleeve connected to the cross head for turning the blades. Sleeve sliding on the shaft inside the stern tube.
⑭ Thrust bearing.
⑮ Non-rotating collar for moving the sliding sleeve.
⑯ Collar thrust ball bearings.
⑰ Guide pin for collar.
⑱ Fork for moving non-rotating collar.
⑲ Links connecting fork with non-rotating collar.
⑳ Bolts for tying the two propeller hub halves together. Propeller hub split in the centers of the propeller bearings.

cylinder types. The smallest is an 8 HP. engine at 900 RPM. and the largest 150 HP. at 300 RPM. All Bolinder oil engines are provided with pre-combustion chambers, are low compression, 2-cycle, and use crank-case scavenging. Because the engines are of low compression type, pre-heating is required. This is done by means of an electric coil before starting, or in some fishing vessels a blow torch is used. The pre-heating requires about 30 seconds. This type of engine has a long life, low maintenance cost, is simple to operate and is preferred by the fishing folk in Denmark, Sweden, and Norway. In fact, about 80 per cent of all fishing vessels in the Scandinavian countries use this type of engine with reversible propellers in sizes up to 150 HP. The ahead and astern movement of the vessel is accomplished by altering the position of the propeller blades in relation to the shaft and any desired speed between zero and the maximum either ahead or astern can be obtained by turning the hand wheel, which is mounted on the top of the maneuvering column. When changing the position of the propeller blades the friction clutch connecting the engine to the propeller must be disengaged because the forces, when turning the

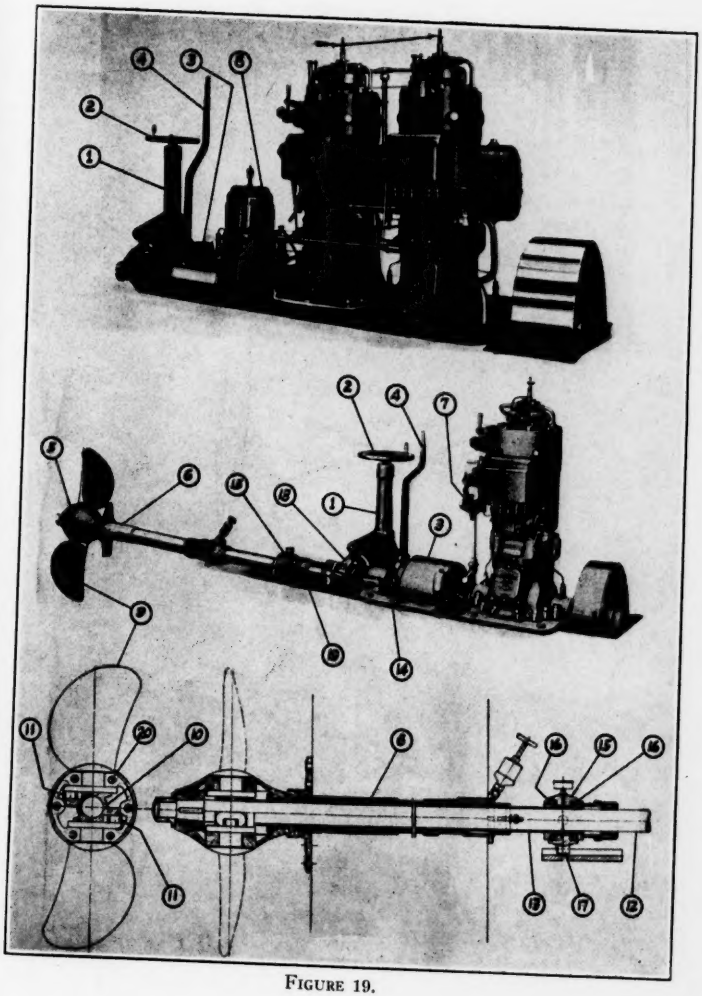
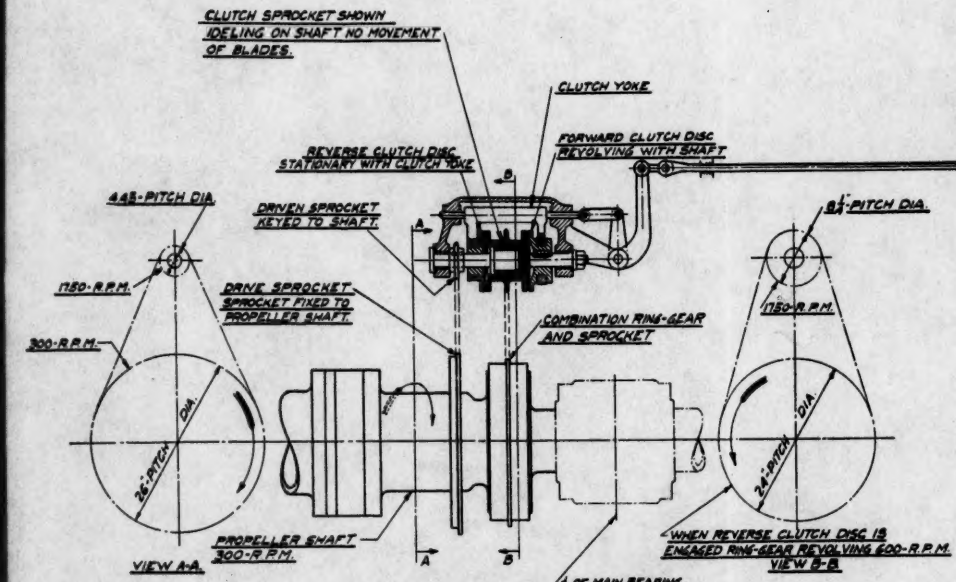
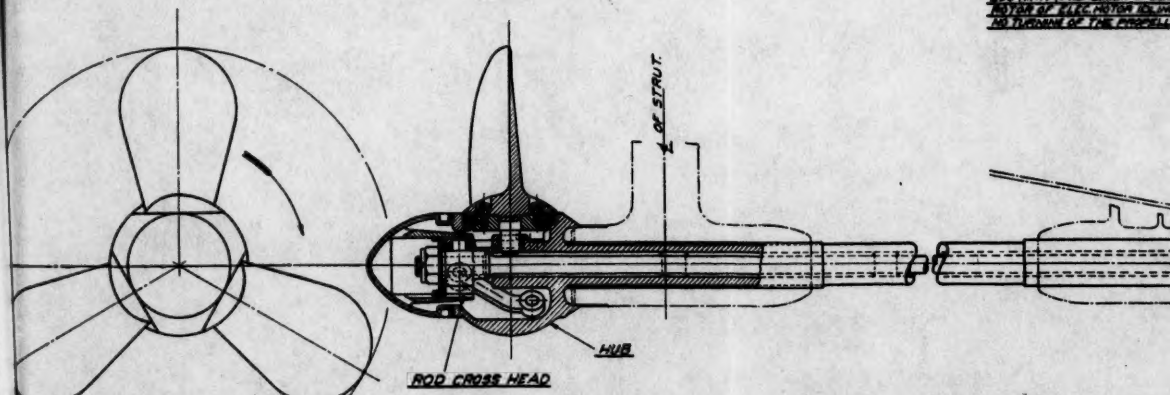


FIGURE 19.


CLUTCH SPROCKET SHOWN IDELING ON SHAFT NO MOVEMENT OF BLADES.
CLUTCH YOKE
REVERSE CLUTCH DISC STATIONARY WITH CLUTCH YOKE
FORWARD CLUTCH DISC REVOLVING WITH SHAFT
POSITION OF LEVER WHEN REVERSE CLUTCH DISC IS ENGAGED AND BLADES TURNING FROM FULL FORWA PITCH TO FULL REVERSE PIT
44½-PITCH DIA.
DRIVEN SPROCKET KEYED TO SHAFT.
8½-PITCH DIA.
1150-R.P.M.
DRIVE SPROCKET SPROCKET FIXED TO PROPELLER SHAFT.
COMBINATION RING-GEAR AND SPROCKET
300-R.P.M.
1150-R.P.M.
26"-PITCH DIA.
24"-PITCH DIA.
PROPELLER SHAFT 300-R.P.M.
WHEN REVERSE CLUTCH DISC IS ENGAGED RING-GEAR REVOLVING 600-R.P.M.
VIEW A-A.
VIEW B-B
℄ OF MAIN BEARING FOR PROPELLER SHAFT.
300-R.P.M.
THE WHOLE PITCH REVOLVIN 300-R.P.M. PROPELLER SPEE ROTOR OF ELEC-MOTOR IS NO TURBINE OF THE PROPEL

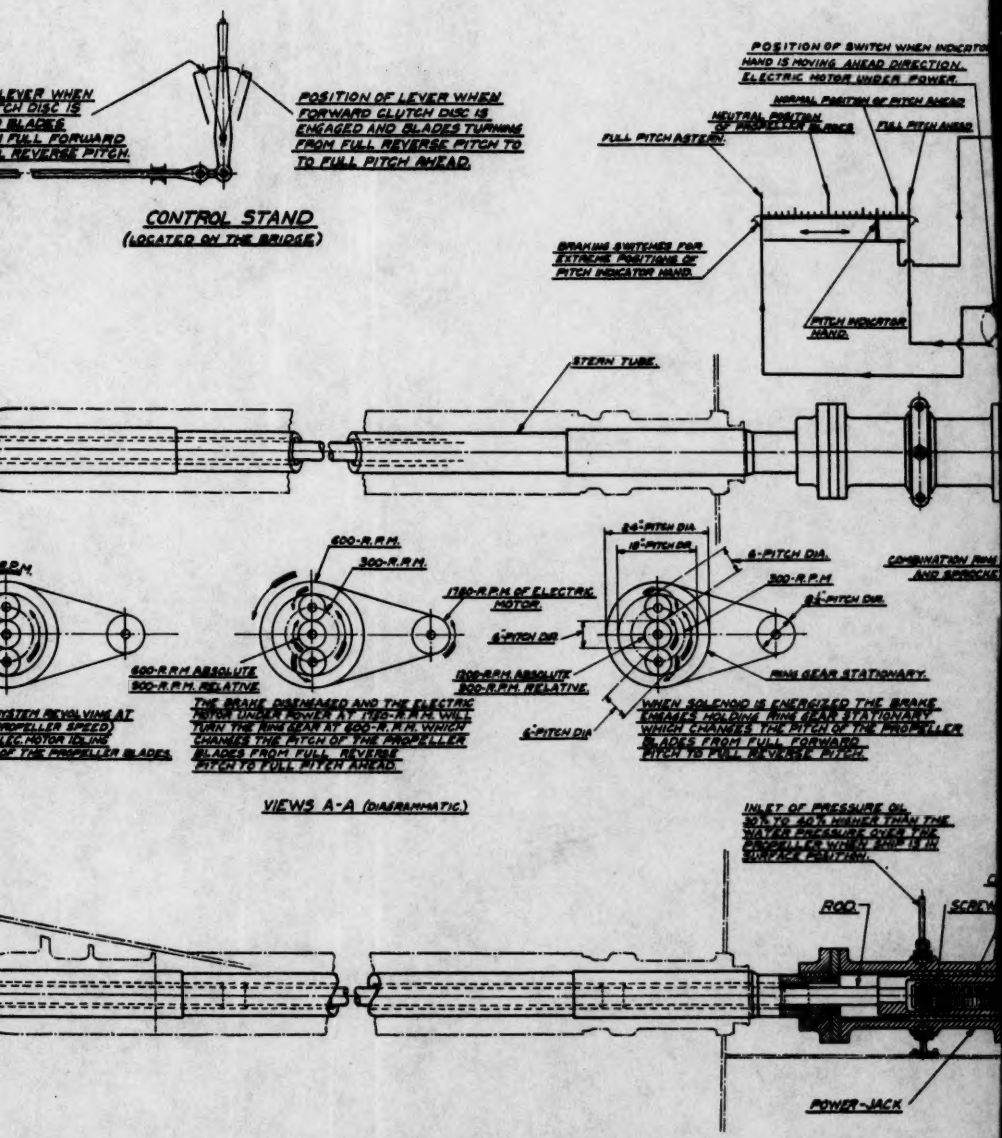
PROPELLER SHAFT DRIVE.

APPLICABLE ONLY WHEN THE DISTANCE TO THE BRIDGE PERMITS MECHANICAL OPERATION OF THE CLUTCHES.



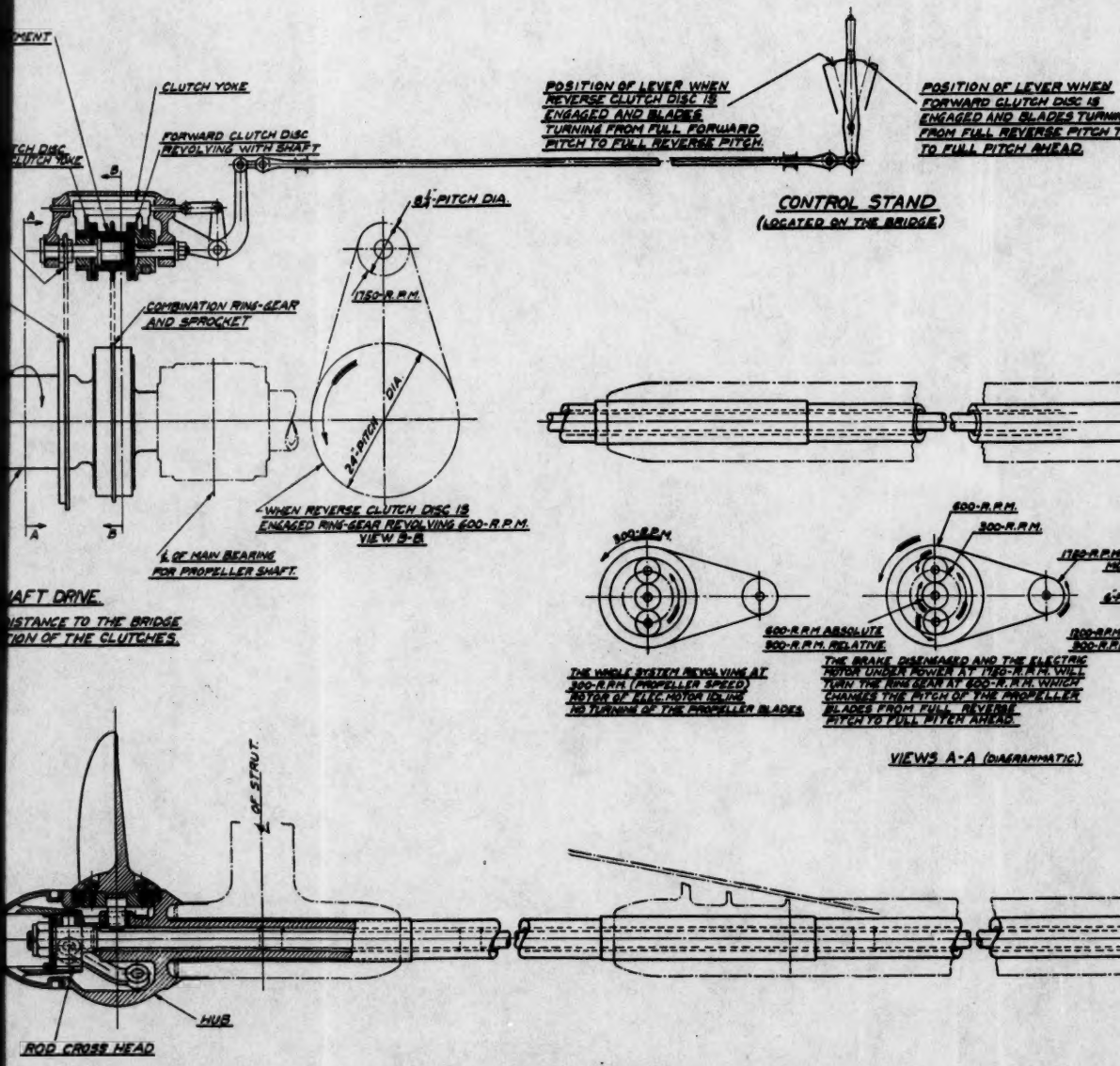
ASSEMBLY PLAN OF CONTROLLABLE PITC

FIGUR


POSITION OF LEVER WHEN FORWARD CLUTCH DISC IS ENGAGED AND BLADES TURNING FROM FULL REVERSE PITCH TO TO FULL PITCH AHEAD.
CONTROL STAND
(LOCATED ON THE BRIDGE)
POSITION OF SWITCH WHEN INDICATOR HAND IS MOVING AHEAD DIRECTION. ELECTRIC MOTOR UNDER POWER.
NORMAL POSITION OF PITCH AHEAD
NEUTRAL POSITION OF PROPELLER BLADES
FULL PITCH ASTERN
FULL PITCH AHEAD
BRAKING SWITCHES FOR EXTREME POSITIONS OF PITCH INDICATOR HAND.
PITCH INDICATOR HAND.
STERN TUBE.
600-R.P.M.
300-R.P.M.
1780-R.P.M. OF ELECTRIC MOTOR.
24"-PITCH DIA.
18"-PITCH DIA.
6"-PITCH DIA.
300-R.P.M.
8½"-PITCH DIA.
6"-PITCH DIA
600-R.P.M ABSOLUTE
300-R.P.M. RELATIVE.
1186-R.P.M. ABSOLUTE
300-R.P.M. RELATIVE.
6"-PITCH DIA
RING GEAR STATIONARY.
THE BRAKE DISENGAGED AND THE ELECTRIC MOTOR UNDER POWER AT 1780-R.P.M. WILL TURN THE RING GEAR AT 600-R.P.M. WHICH CHANGES THE PITCH OF THE PROPELLER BLADES FROM FULL REVERSE PITCH TO FULL PITCH AHEAD.
WHEN SOLENOID IS ENERGIZED THE BRAKE RELEASES HOLDING RING GEAR STATIONARY WHICH CHANGES THE PITCH OF THE PROPELLER BLADES FROM FULL FORWARD PITCH TO FULL REVERSE PITCH.
VIEWS A-A (DIAGRAMMATIC.)
INLET OF PRESSURE OIL 30 TO 40 % HIGHER THAN THE WATER PRESSURE OVER THE PROPELLER WHEN SHIP IS IN SURFACE POSITION.
ROD
POWER-JACK

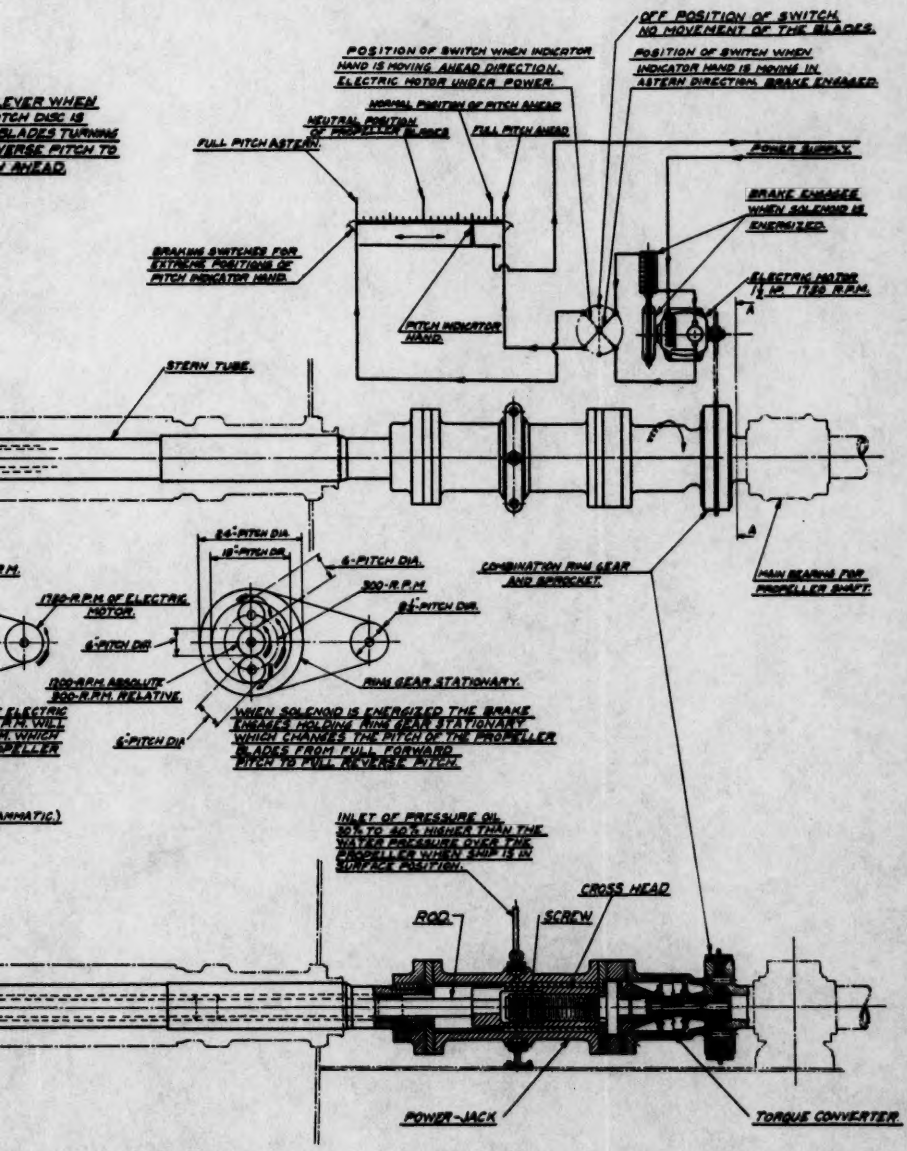
E PITCH PROPELLER ADAPTABLE TO SUBMARINES.

FIGURE 20.

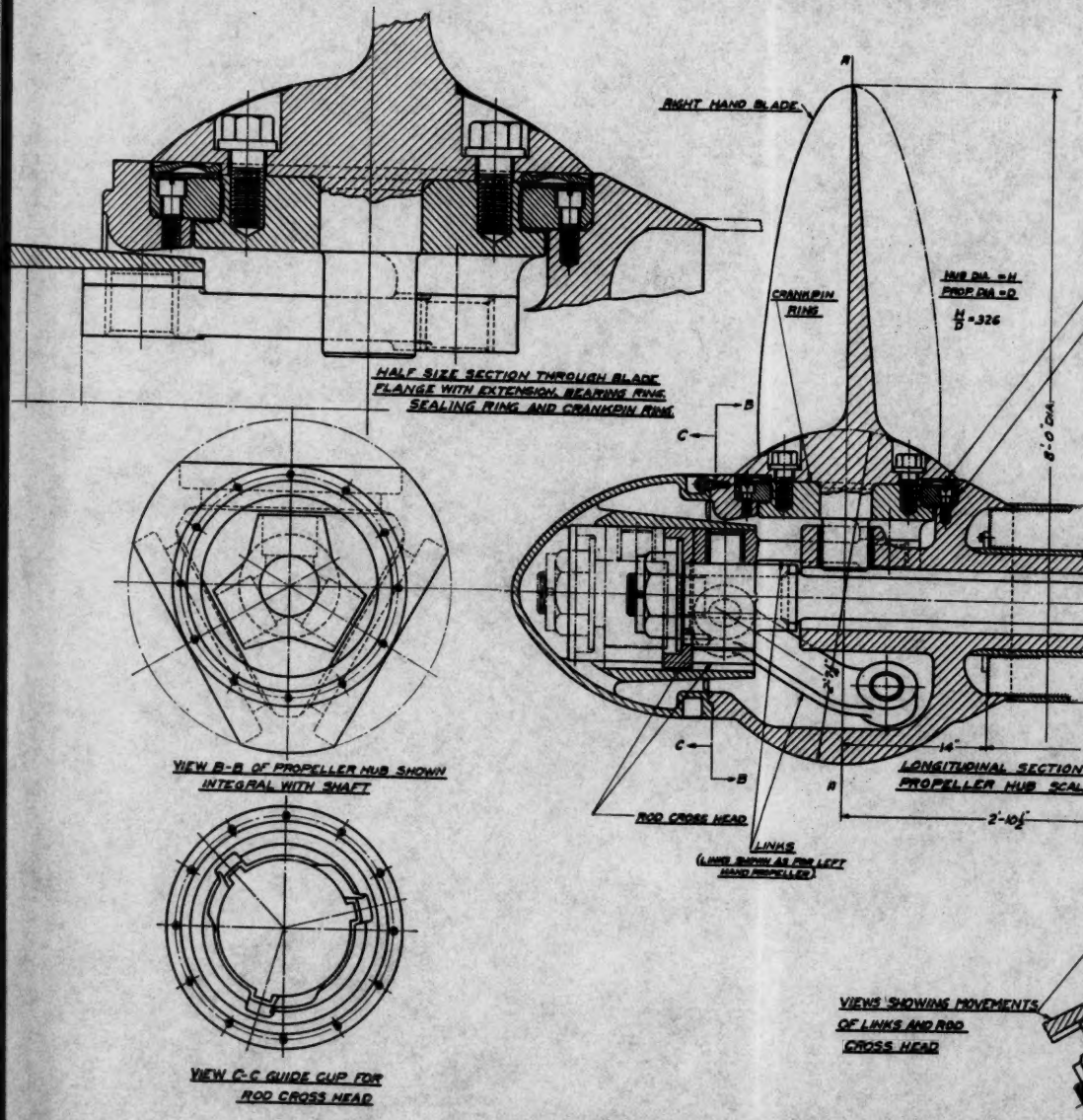

CLUTCH YOKE
FORWARD CLUTCH DISC REVOLVING WITH SHAFT
POSITION OF LEVER WHEN REVERSE CLUTCH DISC IS ENGAGED AND BLADES TURNING FROM FULL FORWARD PITCH TO FULL REVERSE PITCH.
POSITION OF LEVER WHEN FORWARD CLUTCH DISC IS ENGAGED AND BLADES TURNING FROM FULL REVERSE PITCH TO FULL PITCH AHEAD.
CONTROL STAND
(LOCATED ON THE BRIDGE)
8½" PITCH DIA.
1150 R.P.M.
COMBINATION RING-GEAR AND SPROCKET
24" PITCH DIA.
WHEN REVERSE CLUTCH DISC IS ENGAGED RING-GEAR REVOLVING 600-R.P.M. VIEW B-B.
℄ OF MAIN BEARING FOR PROPELLER SHAFT.
A
B
AFT DRIVE.
ISTANCE TO THE BRIDGE
TION OF THE CLUTCHES.
300 R.P.M.
600 R.P.M.
300 R.P.M.
600 R.P.M. ABSOLUTE
300 R.P.M. RELATIVE
THE WHOLE SYSTEM REVOLVING AT 300-R.P.M. (PROPELLER SPEED) MOTOR OF ELEC. MOTOR 150 H.P. NO TURNING OF THE PROPELLER BLADES.
THE BRAKE DISENGAGED AND THE ELECTRIC MOTOR UNDER POWER AT 1150-R.P.M. WILL TURN THE RING GEAR AT 600-R.P.M. WHICH CHANGES THE PITCH OF THE PROPELLER BLADES FROM FULL REVERSE PITCH TO FULL PITCH AHEAD.
VIEWS A-A (DIAGRAMMATIC.)
℄ OF STRUT.
HUB
ROD CROSS HEAD

ASSEMBLY PLAN OF CONTROLLABLE PITCH PROPELLER ADAPTABLE TO

FIGURE 20.

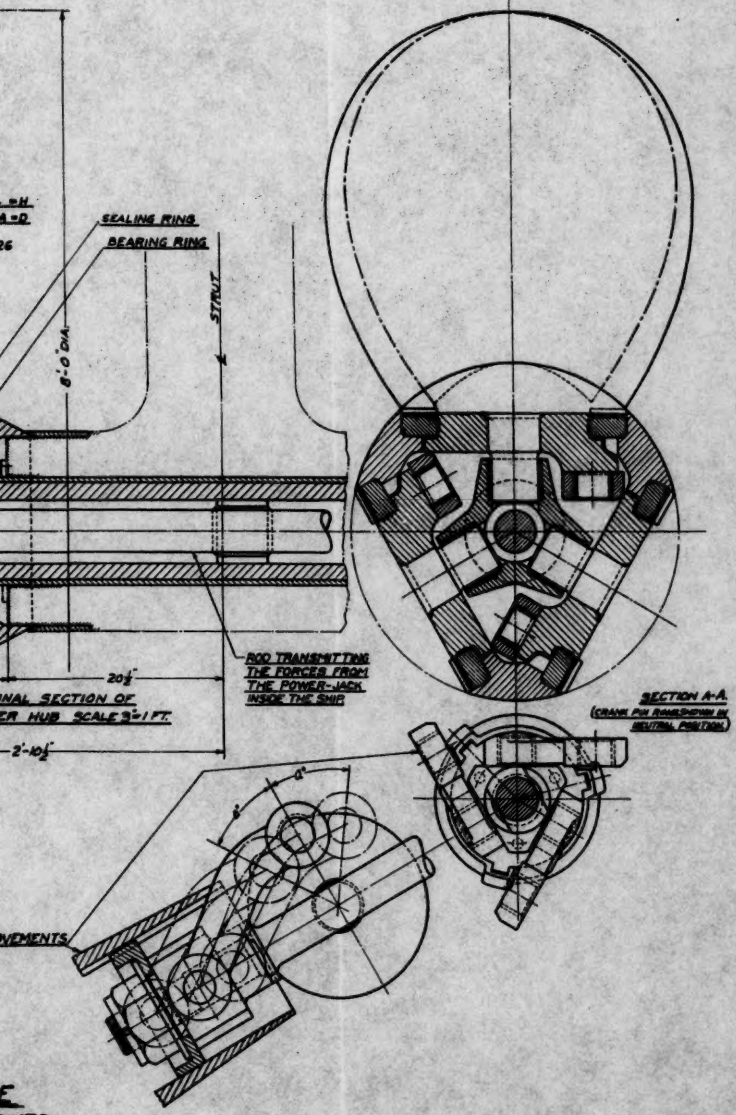

EVER WHEN
TCH DISC IS
BLADES TURNING
VERSE PITCH TO
AHEAD.
POSITION OF SWITCH WHEN INDICATOR HAND IS MOVING AHEAD DIRECTION. ELECTRIC MOTOR UNDER POWER.
OFF POSITION OF SWITCH. NO MOVEMENT OF THE BLADES.
POSITION OF SWITCH WHEN INDICATOR HAND IS MOVING IN ASTERN DIRECTION. BRAKE ENGAGED.
NORMAL POSITION OF PITCH AHEAD
NEUTRAL POSITION OF PROPELLER BLADES
FULL PITCH ASTERN
FULL PITCH AHEAD
POWER SUPPLY
BRAKE ENGAGED WHEN SOLENOID IS ENERGIZED.
BRAKING SWITCHES FOR EXTREME POSITIONS OF PITCH INDICATOR HAND.
ELECTRIC MOTOR 7½ HP. 1780 R.P.M.
PITCH INDICATOR HAND.
STERN TUBE.
24"-PITCH DIA.
18"-PITCH DIA.
6"-PITCH DIA.
300-R.P.M.
8½"-PITCH DIA.
COMBINATION RING GEAR AND BOSCHET.
MAIN BEARING FOR PROPELLER SHAFT.
1780-R.P.M. OF ELECTRIC MOTOR.
6"-PITCH DIA
1200-R.P.M. ABSOLUTE 300-R.P.M. RELATIVE.
RING GEAR STATIONARY.
WHEN SOLENOID IS ENERGIZED THE BRAKE ENGAGES HOLDING RING GEAR STATIONARY WHICH CHANGES THE PITCH OF THE PROPELLER BLADES FROM FULL FORWARD PITCH TO FULL REVERSE PITCH.
6"-PITCH DIA
INLET OF PRESSURE OIL 30% TO 40% HIGHER THAN THE WATER PRESSURE OVER THE PROPELLER WHEN SHIP IS IN SURFACE POSITION.
CROSS HEAD
ROD
SCREW
POWER-JACK
TORQUE CONVERTER.

ABLE TO SUBMARINES.


RIGHT HAND BLADE
CRANKPIN RING
HUB DIA. = H
PROP. DIA. = D
H/D = .326
HALF SIZE SECTION THROUGH BLADE FLANGE WITH EXTENSION, BEARING RING, SEALING RING AND CRANKPIN RING
VIEW B-B OF PROPELLER HUB SHOWN INTEGRAL WITH SHAFT
VIEW C-C GUIDE CUP FOR ROD CROSS HEAD
ROD CROSS HEAD
LINKS
LONGITUDINAL SECTION
PROPELLER HUB SCAL
VIEWS SHOWING MOVEMENTS OF LINKS AND ROD CROSS HEAD

PROPOSED HUB DESIGN OF CONTROLLABLE PITCH PROPELLER ADAPTABLE TO SUBMARINES

FIGURE 21.



...INAL SECTION OF
...ER HUB SCALE 3"=1 FT.

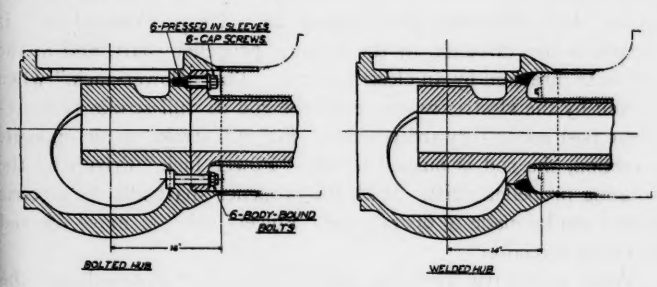
...VEMENTS

...E
...INES

blades against the thrust of the water, are too great to permit hand operation of the reversing mechanism. The maneuvering of a fishing vessel is usually done from the deck cabin. The turning of the propeller blades from the deck cabin is accomplished by a hand wheel which is connected to the shaft of the maneuvering column by means of a system of rods, universal joints and bevel gears. The friction clutch must also, by means of a system of links and levers, be operated from the deck cabin. These engines, with reversible propellers, are specially suitable for fishing vessels, particularly those engaged in trawling and seine netting where the speed must vary within wide limits. When under sail alone the propeller blades can be set so as to offer the least possible resistance.

Proposed Controllable Pitch Propeller Adaptable to Submarines and Also Larger Ships.

The design shown in Figures 20, 21, 22, 23, and 24 is the result of a thorough study of a great number of controllable pitch propeller designs used for hydraulic turbines, ships and airplanes including hydraulically as well as mechanically actuated blade movements.



PROPELLER HUB

Figure 22.

The working principle of the mechanism as shown in Figure 20 is believed to be the simplest possible. The propeller blades which turn in the hub are provided with crank pins which are moved back and forth by a rod located inside the hollow propeller shaft.

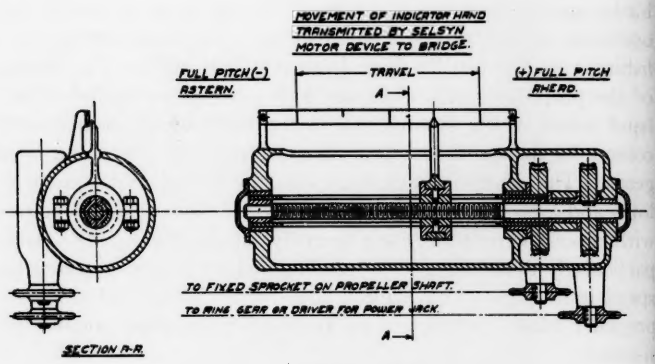


FIGURE 24.

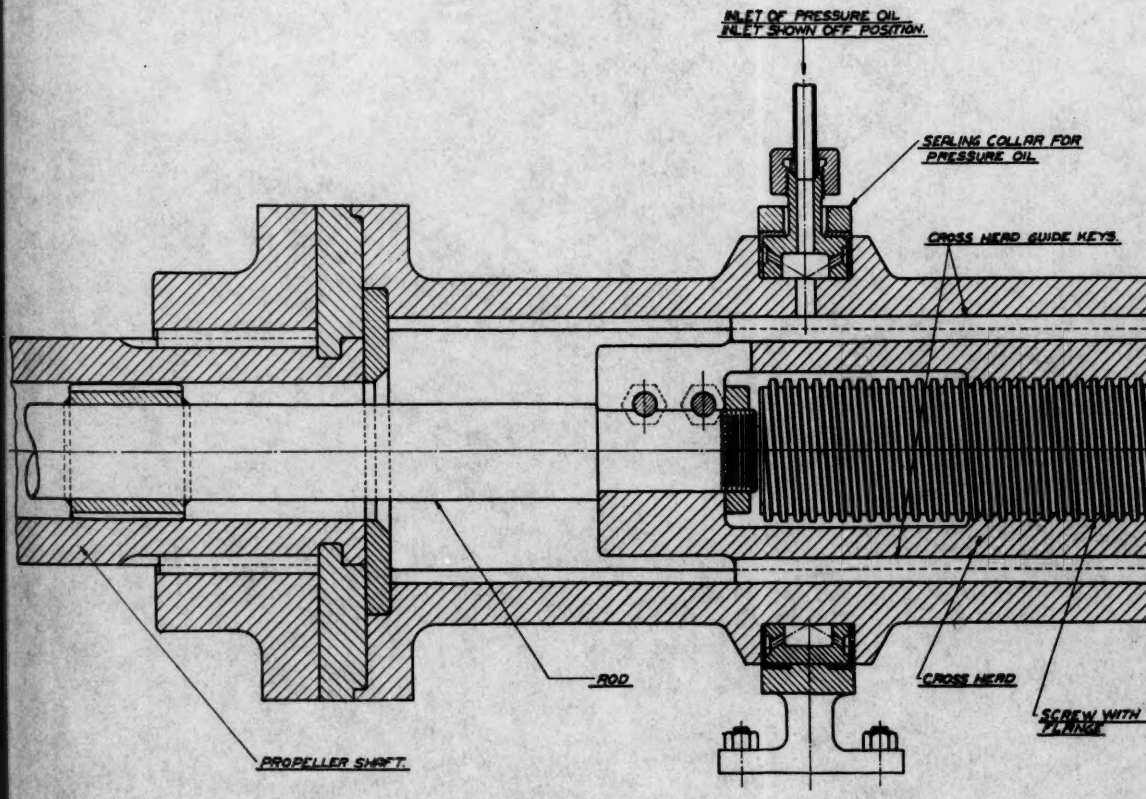
The angular movement of the crank pins is taken up by links connected to a cross head screwed to the end of the rod and guided in the hub. The forces necessary for moving the rod back and forth are produced by means of a " Power-Jack " located inside the propeller shaft, inboard. See Figure 23. This " Power-Jack " is driven in one direction by the rotating propeller shaft, and in the opposite direction by an electric motor. The " Power-Jack," when driven by the electric motor, pulls the rod its full stroke from reverse position to forward position. For full stroke in the opposite direction, the rod is pushed by the " Power-Jack," driven by the rotating propeller shaft. With the propeller shaft idle the electric motor can be made to operate the " Power-Jack " to move the rod in either direction.

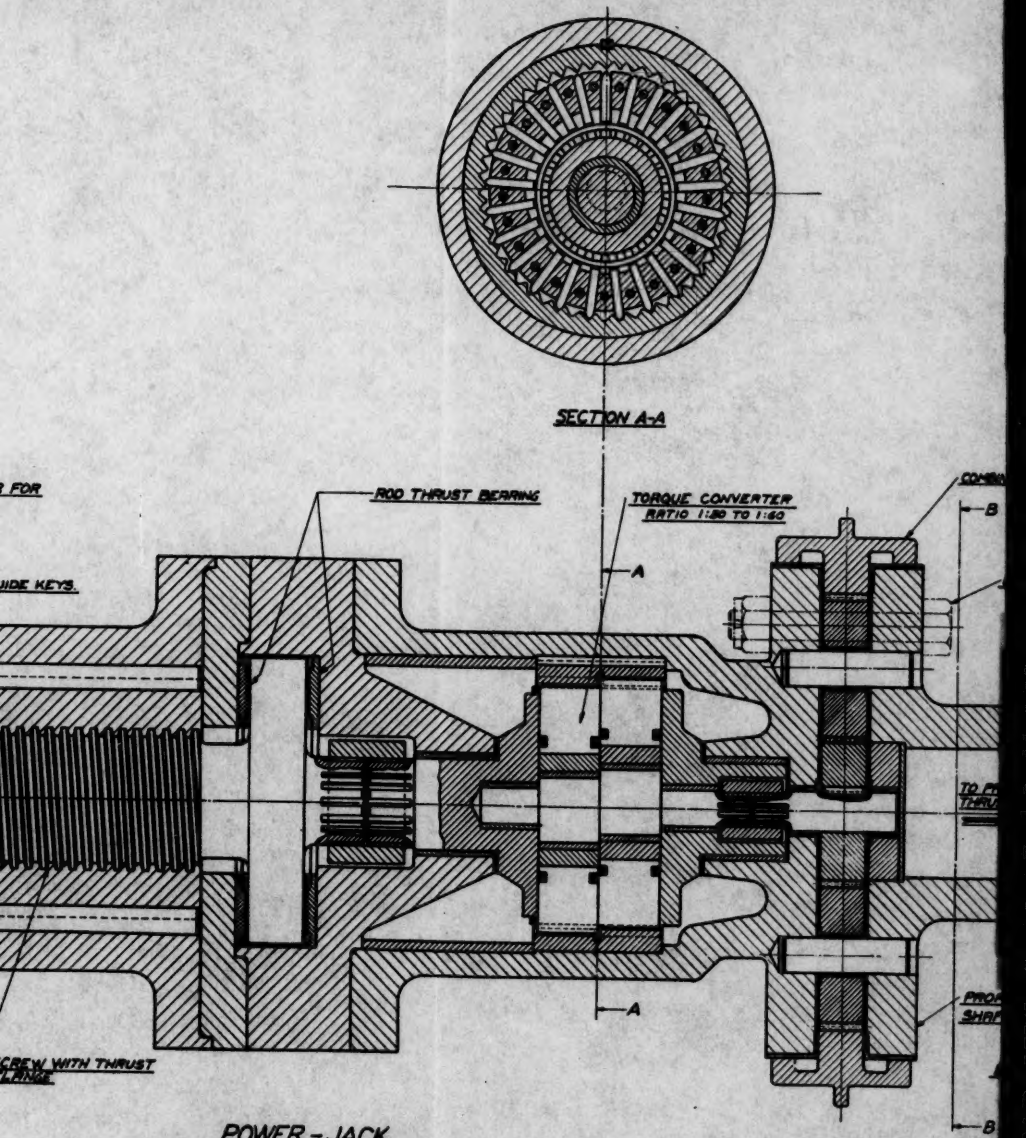
When an electric motor is used to drive the " Power-Jack " the arrangement is as follows: The end opposite the driving end of the motor-rotor shaft is attached to a brake-drum. The brake shoes are operated by a solenoid electrically connected with the pitch indicator hand and the electric motor in such a way that the motor and the solenoid are operated alternately depending on whether the position of the switch requires forward or aft movements of the rod connecting the cranks of the propeller blades.

PITCH

inks con-
guided in
and forth
e the pro-
Jack" is
nd in the
k," when
from re-
e opposite
en by the
ne electric
e the rod

Jack" the
ng end of
The brake
with the
y that the
nding on
aft move-
blades.

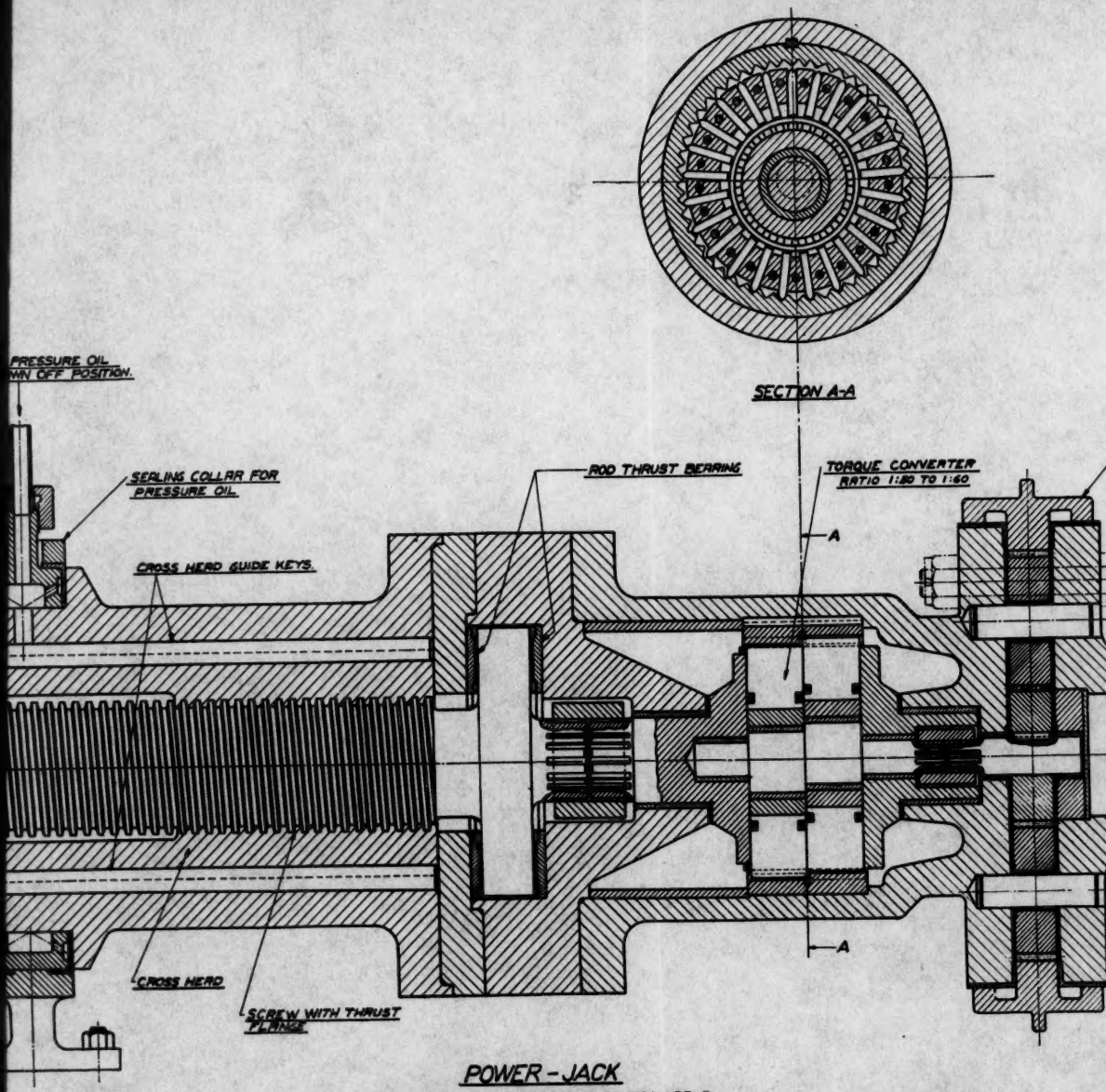

INLET OF PRESSURE OIL
INLET SHOWN OFF POSITION.
SEALING COLLAR FOR PRESSURE OIL.
CROSS HEAD GUIDE KEYS.
ROD
CROSS HEAD
SCREW WITH FLANGE
PROPELLER SHAFT.



POWER-JACK

FOR TURNING THE BLADES IN A CONTROLLABLE PITCH PROPELLER ADAPTABLE TO SUBMARINES

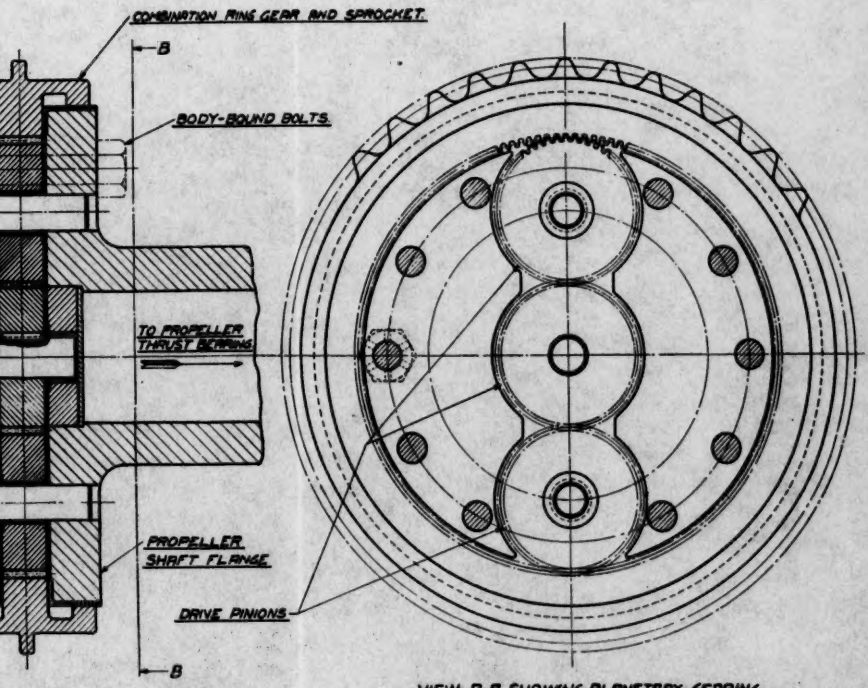
FIGURE 23.



POWER-JACK

FOR TURNING THE BLADES IN A CONTROLLABLE PITCH PROPELLER ADAPTABLE TO SUBMARINES

FIGURE 23.



VIEW B-B SHOWING PLANETARY GEARING WITH PROPELLER SHAFT FLANGE REMOVED.

The electrical hook-up is provided with contact breaking switches at the extreme positions of the indicator hand in order to prevent the " Power-Jack " from accidentally breaking any part in the hub.

As shown in upper left-hand corner of Figure 20, the " Power-Jack " may also be operated in either direction entirely by the rotating propeller shaft. The transmission as pictured consists of a shaft with a keyed-on sprocket driven by the propeller shaft, and one idling sprocket connected to the combination ring gear and sprocket driving the " Power-Jack " and provided with two clutch discs. These clutch discs are alternately engaged either by a clutch disc fixed to a yoke which cannot rotate about the shaft and can move laterally, or by a clutch disc movable with the same yoke and sliding in a keyway, and revolve with the shaft. The clutches are mechanically operated from the bridge.

A pitch indicator mechanism is shown in Figure 24. However, a number of different designs can be developed to suit each particular installation. This type of indicator is mentioned because it can be made very compact and reliable and has a general application for this type of mechanical drive. This indicator is based on the relative rotation between the propeller shaft and the combination ring gear and sprocket driving the " Power-Jack." Differences in rotation between the two members indicates a corresponding movement of the propeller blades. The design consists of two independent sets of worm and gear reducer driven by chains from the propeller shaft and ring gear, respectively, and working in the same housing with output shafts arranged concentrically with each other. One shaft is threaded and rotates inside the other shaft which consists of a steel tubing provided with long slots. The screw shaft carries a nut-type crosshead sliding in the slots of the steel tubing shaft. With the same rotation of the worm and gear reducers the crosshead rotates with the two shafts without axial movements. Any difference of rotation will move the crosshead axially in one or the other direction. The axial movements of the crosshead which indicates the position of the propeller blades may be transmitted to the bridge, by means of a mechanical arrangement or by electrical means, such as Selsyn motor devices for larger ships.

It has been found satisfactory in Kaplan turbines to use a slight oil pressure inside the hub in order to prevent water seepage through the seals at the blade bearings as an additional safeguard against water penetrating into the hub. The same method is used in several controllable pitch propeller designs and it is believed to be quite essential for the purpose of lubricating the mechanism and keeping the water out of the propeller hub. In a propeller for a ship we are confronted with a sealing problem which requires much more attention because we have to deal with salt water. The Kaplan turbines have completely overcome the sealing difficulties, and although they work in fresh water, we may benefit from their experience. As indicated in Figures 20 and 23 an oil inlet ring has been provided for the "Power-Jack" to permit oil to be led into the shafting system. For a surface vessel a tank above the water line may be located somewhere in the ship and piped to the oil inlet ring. A glass gage on the tank will permit occasional observation of the oil level. Thus, a check on the wear in the bearings and sealing rings may be observed. In a submarine installation, a tube line from the discharge side of the lubricating oil pump attached to the engine may be connected to the oil inlet ring. The oil line should also be provided with a check valve, a reducing valve and a flowmeter for measuring the small quantities of oil used. This arrangement may take the place of the pressure tank described above.

Mechanical drives have been tried in the past and found less satisfactory than hydraulic drives. This has been due to the fact that a compact, yet strong enough gear arrangement has not been employed. The writer believes that the difficulties encountered in the previously used mechanical drives will be entirely overcome by introducing the device shown in Figures 20, 23 and 25. This device, the "Heliocentric Reducer" as named by the manufacturer, is very extensively used in industry today in various applications and in some instances works under much more severe conditions than it ever will be called upon to meet when used in a controllable pitch propeller installation for a ship.

The hydraulic drive has been criticized for not being able to maintain a certain desired fixed pitch of the blades when the propeller is subjected to the full thrust. If such condition has actually

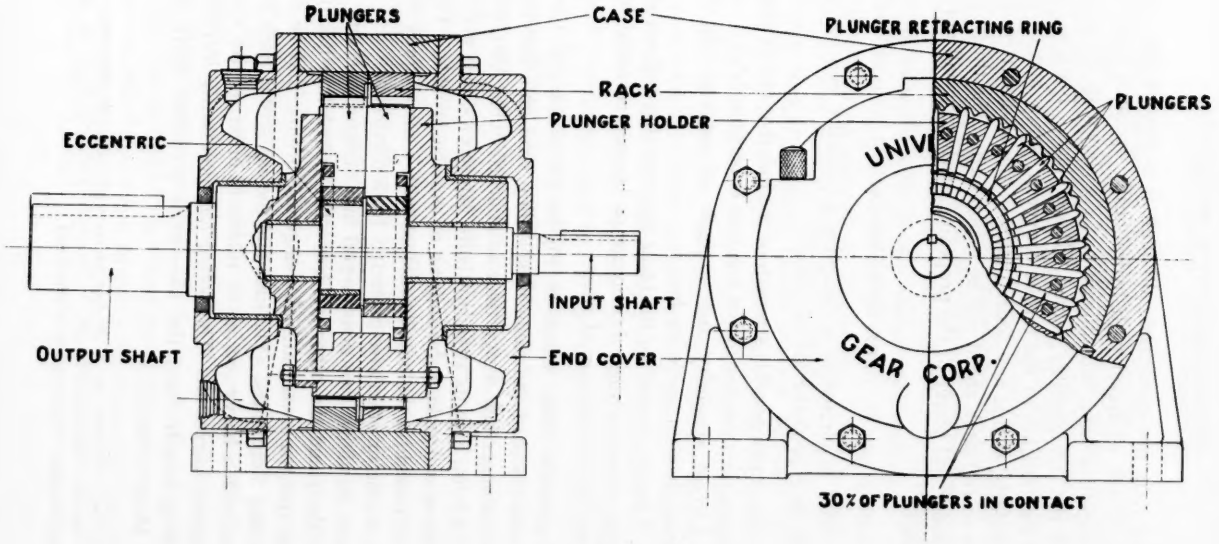

PLUNGERS
CASE
PLUNGER RETRACTING RING
RACK
PLUNGER HOLDER
PLUNGERS
ECCENTRIC
UNIV
INPUT SHAFT
OUTPUT SHAFT
END COVER
GEAR CORP.
30% OF PLUNGERS IN CONTACT

FIGURE 25.

occurred it is not confirmed, although it is known that certain foreign vessels equipped with hydraulically operated controllable pitch propellers have had difficulties. As shown in the arrangement of the sub-assembly named "Power-Jack," Figures 20 and 23, the combination of the torque converter, screw and crosshead is self-locking, so that any desired pitch of the blades from neutral position to full pitch ahead will be maintained. The thrust of the water under full power of the propelling machinery will not move the blades.

The Hub Design.

The chief requirements in designing a propeller hub for a controllable pitch propeller are as follows:

(*a*) Maximum possible strength of the hub that can be obtained considering the diameter of the hub that is permitted.*

(*b*) Utmost simplicity and strength of parts of the blade-turning-mechanism inside the hub.

(*c*) A sealing device for the blade bearings that will not change its spring characteristics, deteriorate or become corroded by salt water and has a minimum of wear.

The propeller hub, shown in Figure 21, is integral with the shaft and made of the same nickel alloy steel and thereby produces a stronger hub design than an attached hub. Also, elimination of a joint, with its attendant bolts, cap screws, nuts and shearing sleeves will reduce the sources of possible trouble. The hub may be treated on the outside surface with a metal spray of cadmium or zinc or otherwise protected from corrosion.

Figure 22 shows two non-integral hub designs, one hub bolted to the shaft and the other welded. The bolted hub has the advantage that the same material, manganese bronze, may be used as in the propeller blades. The nuts and the cap screw heads may be covered with a heavy coating of rubber in order to give a protection against corrosion. The bolted hub necessarily increases the over-hang length from the after-most propeller shaft bearing about 15 per cent.

* (Hub dia. = Propeller dia. × .35 does not appreciably affect the efficiency of the propeller. See Dr. Nordstrom's article, "Controllable Pitch Propellers," in the Naval Engineer's Journal, February issue of this year.)

The shaft with welded hub, as shown in Figure 22, will have the same length of over-hang from the after-most propeller shaft bearing as the integrally forged shaft, and the cost of manufacturing will be considerably lower. If the welded construction is to be used it will be necessary to control the many variables which affect the quality of the weld, particularly the material, the joint design, the electrodes, and preheating and postheating treatments. The use of welding on heat-treated alloy steels is somewhat unconventional and should be approached with caution.

The "Bearing Rings" shown in Figure 21 are screwed to the hub with a large number of cap screws. The propeller blade flanges screwed to the "Crank Pin Rings" enclose parts of the "Bearing Rings" creating liberal bearing surfaces and sturdy bearing supports for the propeller blades. Each blade is made integral with a stub shaft which has its support in the center part of the hub. Between the "Bearing Rings" and the propeller blade flanges are sprung flexible metal "Sealing Rings." The "Sealing Rings" move with the blade flanges and slide against the "Bearing Rings." Practically the same principle in design is used, and has been successfully used for the last 16 years, in a great number of large Kaplan turbines throughout the world. On top of the "Bearing Rings" are placed babbit faced aluminum bronze rings to cover the cap screw holes, as well as to give suitable bearing surfaces for the "Sealing Rings." The surfaces of the "Crank Pin Rings" rubbing against the "Bearing Rings" are babbited. The material of the "Bearing Rings" should be nickel alloy steel. The "Crank Pin Rings" are of the same importance as the "Bearing Rings," requiring maximum strength. They, therefore, should be of the same material and heat-treatment. The pins should be given hard surfaces to give a minimum of wear in order to keep the variations in clearances of the link bearings at a minimum. The "Links" should be made of steel and designed for greatest strength within the limited space allowed for their operations. Hard bronze bushings should be provided for the link bearings. The design of the "Rod Cross Head" screwed to the hub-end of the "Rod" and carrying the link-ends opposite the crank-pin ends should be of the same heat-treated material and strength as the "Links." The "Rod Cross Head" pins should be

given the same surface hardness as the crank pins. The long "Rod" connecting the two cross heads in the hub and the "Power-Jack" inside the ship should have sufficient diameter and should be of the proper material characteristics to keep the elongation of the material to a minimum. To prevent buckling of the "Rod" it should be provided at appropriate points with guide bushings of valve bronze brazed to the rod. The "Guide Cup" guiding the "Rod Cross Head" and, also, taking the twist of the "Rod Cross Head" induced by the link motion would be sufficiently strong if cast of valve bronze. The material and the shape of the "Sealing Rings" have to be given particular consideration due to their important functions. They should not only automatically follow the tipping action of the blades when the blades are moving, but must also deflect and increase their sealing properties for variations in water pressure (as in submarine installations); be corrosion resistant to salt water and have good wearing qualities. It is believed that a copper beryllium alloy or a chromium-nickel alloy might offer the solution as these alloys are known to be a corrosion resistant, and at the same time have very excellent spring qualities. The propeller with its component parts is symmetrically arranged in respect to the rotating axis. Each part has a simple form and can be made to exact dimensions and weight distribution relative to its counterparts. The propeller blade flange is provided with slotted holes to take the cap screws connecting the blades to the "Crank Pin Ring." Each blade can be correctly adjusted in relation to the other blades by means of the slotted holes. After the exact position of the blades is obtained, the blades are doweled in place. Due to the symmetry and simplicity of parts, statical and dynamical balancing to sufficiently accurate degree will not be difficult.

The "Power-Jack" Design.

The "Power-Jack" design, Figure 23, owes its compactness and simplicity to the use of the Heliocentric reducer. Figure 25 shows in section a typical conventional reducer of the Heliocentric type. The internal assembly of this unit is identical to the one used in the "Power-Jack" and is named "Torque Converter."

Heliocentric gears have been used for a great number of different applications. Approximately 50,000 units have been put out

in the last ten years. The majority of these units have been furnished in connection with applications requiring a high ratio of reduction where the Heliocentric gear offers a combination of maximum efficiency and maximum reliability. Hundreds of gears of other types in practically all fields have been replaced by this type.

The application which offers the greatest similarity to the propeller pitch changing service is the application of the Heliocentric gear in connection with handling of inlet vanes to hydraulic turbines. A few of these installations have been made and they have proven universally successful. It will be understood that for an application of this kind the greatest reliability is paramount since the torque converter forms a most vital link between the governor and the control of the water to the turbines. This Heliocentric reducer must have the greatest possible reliability since the entire performance of the unit depends on it.

Another application which is somewhat similar is the use of the Heliocentric gear in connection with valve operators for city water plants and sewage disposal plants. Here, also, maximum reliability is demanded since some of these drums stand still for long periods and are suddenly called upon to perform their work. A failure to do so might vitally upset the entire operation of a city water distribution system.

Principles of Operation.

The Heliocentric reduction design requires but two principal moving elements. First, an input shaft carrying a centrally located eccentric. And second, an output shaft attached to a plunger holder which revolves within a stationary rack at the rate of one rack tooth for each revolution of the input shaft.

The plunger holder is slotted radially to carry a battery of reciprocating plungers working into and out of the stationary rack. This rack is keyed to the case and has the general appearance of an internal ring gear but with a definitely different tooth contour.

As the input eccentric rotates, the plungers work into and out of the stationary rack, producing a combination of sliding and rolling wedge action which forces rotation of the output shaft assembly with a minimum of friction.

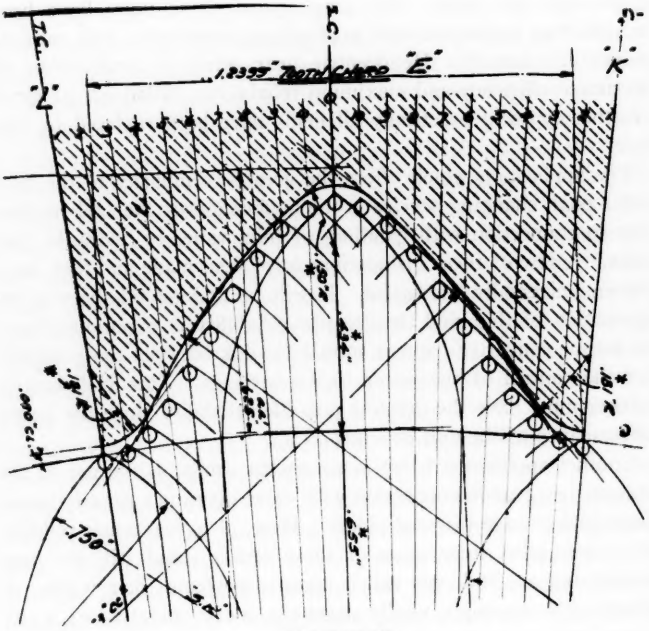


DIAGRAM 26.

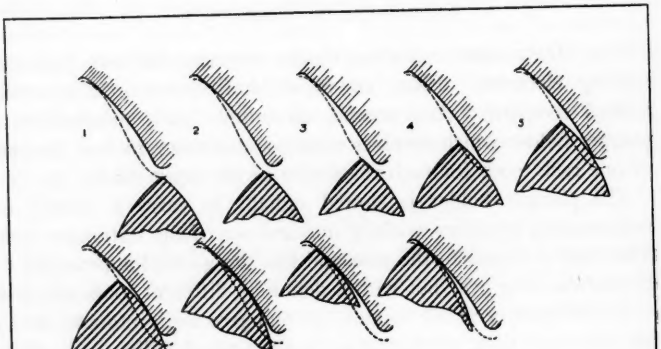


DIAGRAM 27.

It may be noted from the diagrams 26 and 27 that the eccentric on the input shaft of the Heliocentric reducer forces its plungers outward and into the stationary rack. The contour of the rack teeth impose a uniform angular motion. Since the part holding the plunger is rigidly connected to the output shaft, a smooth uniform rotation results. The plungers come into engagement progressively and as each reaches the limit of its stroke, it is retracted by the ring working in unison with its eccentric.

By following the action of one plunger, it will be noted that for one complete revolution of the eccentric, the plunger has advanced a distance equivalent to one tooth of the rack. Another revolution of the eccentric (high speed shaft) will cause the plunger with its holder or retainer (slow speed shaft) to advance another tooth, and so on. Thus it will be seen that the speed reduction ratio is equal to the number of teeth in the rack *plus one* when the input and the output shafts rotate in the same direction. For example, in a 30 to 1 reducer with both input and output revolving in the same direction, there are 15 plungers and 29 teeth in the internal gear rack. However, the output shaft may just as readily be made to rotate in the opposite direction from the input. Example of 30 to 1 ratio with opposite rotations: 15 plungers and 31 teeth in the internal gear rack.

Actually the plunger action in a Heliocentric reducer is not a simple sliding or wedge action. The contact is a combined sliding and rolling action. The difference is important and is illustrated diagrammatically in the charts, Figures 26 and 27. The motion and contacts of a single plunger are shown throughout a full cycle of engagement with the rack that is keyed to the reducer case.

The plunger point follows a double sine curve while the contact starts on the radius or shoulder of the plunger and gradually advances to take the load on its point. This action avoids plain sliding friction by producing a combined rolling-sliding contact which progressively rolls the lubricant in at the point of greatest pressure. This action has been proved by actual tests, and accounts for the high efficiencies obtained on high ratios.

Closely related to the above is another factor not immediately obvious. Refer to the diagrams once more and note that the plunger extends part way into the tooth space before making contact. This, however, is a variable factor and the variations work

favorably so far as capacity is concerned. In the diagrams the contacts presented are normal for the unit running idle (without load). As a load is imposed and as the torque increases, slight deflections occur on the contacting faces of the plunger holders so that the rack teeth start the plungers into engagement earlier. This means that as the load increases a greater number of plungers are contacted at the same instant and under test it has been found as many as 9 plungers in contact in an element carrying 24 plungers.

Fully 30 per cent of the plungers are always engaged, thus giving load distribution and making possible a very compact unit without undue strain on individual elements. Contrast this with the ordinary tooth to tooth gear where usually one tooth (seldom more than three teeth) carries the load at any instant.

The remaining parts shown in the "Power-Jack," Figure 23, are of simple construction and present no design difficulties. It is suggested that the "Cross Head" material be Gun bronze. The material of the "Screw with Thrust Flange" may be nickel alloy steel heat-treated. The sides of the thrust flange integral with the screw should be given hard surfaces as they are working against babbitted steel rings. The planetary gears may all be made of alloy steel heat-treated and the teeth given a hard surface for wear. The "Combination Ring Gear and Sprocket" should have the surfaces riding on the bushings hardened. All bearing bushings should be made of hard bronze. The material of the parts in the torque converter is well established from the experience of many years and under various working conditions, so there should be no reason to be skeptical of the reliability of such a unit, especially as the working condition in a propeller installation is intermittent and consequently the wear should be negligible. Where the distance between the hub and the inside of the stern tube is extremely long the "Power-Jack," save the planetary gearing, may be located at and streamlined with the outside diameter of the strut bearing. The torque converter may be driven by a light torque tube properly supported and extending to the planetary gearing at the stern tube inside the ship.

The writer fails to understand where a hydraulic drive offers any advantages over those above discussed in the mechanical drive, when considering the hydraulic drive's simplest form of installa-

tion. For example, consider the case when the propeller blade actuating rod can be extended through the large reduction gear shaft. The servomotor piston is connected to the end of the rod which rotates with the shaft inside the stationary servomotor cylinder. The cylinder is bolted to the gear housing. In a hydraulic drive the following items are needed to take the place of the "Power-Jack."

(*a*) A piston and a cylinder with the attached oil regulating mechanism.

(*b*) A motor driven pressure pump of 700 to 1000 pounds pressure per square inch.

A high pressure pump is by no means a simple and entirely foolproof mechanism. The pump must work continuously in order to maintain a constant oil pressure ready at any desired moment to move the servomotor piston. Any position of the piston, save the full ahead and reverse position, must be maintained by constant movement of the regulating mechanism in order to counteract the water pressure on the propeller blades. The high pressure oil piping will also give another source for possible trouble.

What Advantages Has the Controllable Pitch Propeller for Submarines and Other Ships?

Assume that we have a submarine provided with two controllable pitch propellers, each driven by a 600 BHP. Diesel engine connected to the propellers through a fluid or electromagnetic coupling and reduction gears. Between the coupling and the reduction gears is interposed a D. C. motor-generator used as a motor for propelling the vessel in submerged condition and as a generator for feeding the batteries when the ship is in surface condition using the excess power of the engine when cruising.

In order to give some idea about the advantages gained by using a controllable pitch propeller in connection with the above described engine arrangement, it is necessary to refer to Dr. Nordstrom's chart, Figure 28, made up for a twin screw torpedo boat. Such a chart can equally well be made up for a submarine with known resistance curve and a power plant of the above assumed size. It is noted from the chart that a reduction in the maximum speed of the ship of from 12.5 per cent to 25 per cent raises the

available power for feeding the batteries about 30 per cent to 35 per cent, respectively, as compared with the condition when a fixed propeller is used because the pitch can be changed to a suitable value to maintain full rotative speed of the engine. It will also be recognized from the same chart that the excess power of an overloaded engine can be used more effectively with controllable pitch propeller than with one with a fixed pitch. If the submarine is operated under conditions that demand neither maximum possible

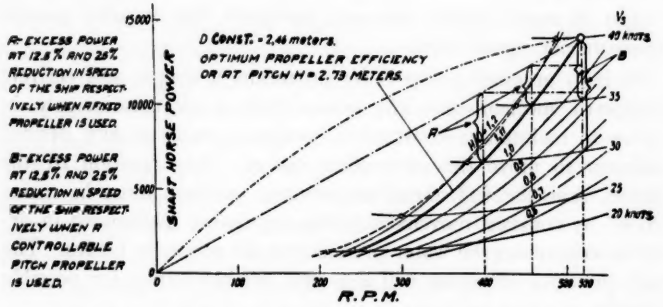


FIGURE 28.—REPRESENTS THE CURVES OF CONSTANT SPEED AND CONSTANT PITCH-RATIO, RESPECTIVELY, AT A GIVEN PROPELLER-DIAMETER AND A GIVEN RESISTANCE CURVE FOR THE SHIP.

The small circles indicate the minimum points on the curves for constant speed. The curves for a larger pitch-ratio than 1.2 are not shown. The family of curves are representative of a case in general. The dash and dotted curves indicate assumed boundary curves for steam turbine and Diesel engine or reciprocating steam engine, respectively.

electrical energy for the batteries, nor the maximum cruising speed, i. e., full rated horsepower of the power plant is not required, it might be desirable to operate the engines under the condition that the lowest fuel consumption per horsepower hour be obtained. Again we point out that the controllable pitch propeller offers considerable possibilities. In order to study closer this question we must know the fuel consumption per horsepower-hour for the whole speed range of the engines. The relation of the characteristic fuel consumption chart of the engine and the chart of the engine-propeller-ship combination of the type shown in Figure 28 is best illustrated by superimposing one of the charts

on the other. From such a combination chart we can evidently obtain at each horsepower value that revolution which corresponds to the lowest fuel consumption per horsepower-hour and, also, get an idea of the corresponding pitch of the propeller. Characteristic fuel consumption curves for a Diesel engine are shown in Figure 29. The chart is plotted for a 4-stroke-cycle engine with natural aspiration. These "profile curves," of course, are expen-

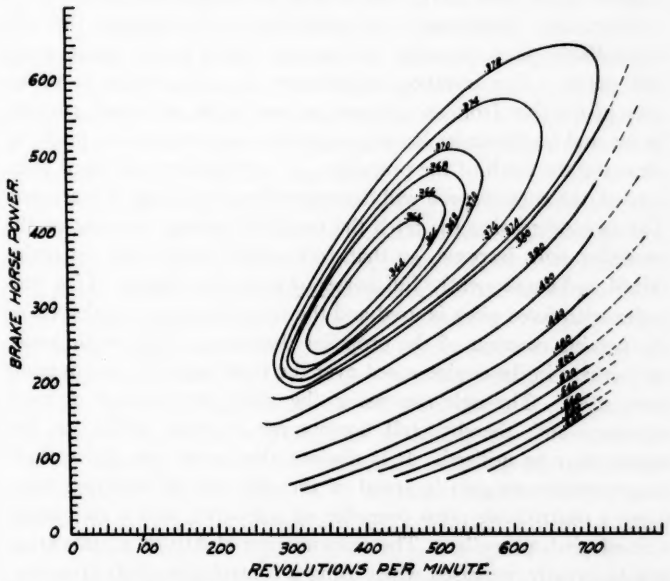


FIGURE 29.—FUEL CONSUMPTION IN POUNDS PER BHP.-HR. AT VARIOUS SPEEDS AND LOADS OF A 6-CYLINDER 12½" X 13" ENGINE THAT DEVELOPS 600-B.H.P. AT 700-RPM.

Furnished by courtesy of the Amer. Loco. Co.

sive to make, but fuel consumption data over a large range of speeds and horsepower carefully obtained and plotted in this manner will give the testing engineer a good idea of his engine regulation. The curves shown on the chart indicate an engine which is certainly perfectly regulated. However, notwithstanding the care taken to show such a perfectly regulated engine in plotting these curves, this engine is recognized as one of the out-

standing 4-stroke-cycle types in this country and has long ago established its high qualities. The manufacturer has placed about 300 in service during the past ten years. About 90 per cent of these are locomotive type, the remainder being divided between stationary engines for generating purposes and streamlined trains. The manufacturer is also super-charging this engine, using the "Büchi" system. By this means the six-cylinder engine is stepped up to 1000 BHP. and is used in streamlined trains.

From Dr. Nordstrom's investigation it also appears that the controllable pitch propeller utilizes the ship's power plant to its full extent. For instance, adjustment for any change in resistance of the ship from such causes as head wind, tail wind, running in ice and variation in loading condition may always be made by changing the pitch of the propeller. If a propeller with fixed pitch is used, the engine speed will decrease when resistance is increased. Through decreasing the pitch the required turning moment of the propeller will decrease, so that full engine speed may be maintained and consequently full horsepower of the engine. If a propeller with fixed pitch is used and the ship's resistance is decreased the turning moment of the propeller decreases which would cause an increase in the engine speed over the rated speed if no governor were used. Through increasing the pitch the turning moment increases and, thereby, full horsepower at rated RPM. of the engine may be utilized. It is obvious that under the above mentioned conditions gain in speed of the ship will be obtained when using a controllable pitch propeller as compared with a ship using a fixed pitch propeller. The maneuvering ability of a submarine will be greatly increased when using a controllable pitch propeller. With a Diesel engine connected to a fixed propeller the speed of the ship can be reduced to only ⅓ or at most ¼ of the maximum speed. With controllable pitch propeller the speed of the ship can be reduced to any speed from maximum to zero while the engine is running full speed, always ready for instantaneous increase to full power. The maneuvers are smooth, without shocks or vibrations and will hardly be noticed by the ship's personnel. The unpleasant feeling of rapid engine accelerations during frequent stops and starts will be entirely absent.

Dr. Nordstrom points out that a ship's acceleration and retardation are much greater when a controllable pitch propeller is used.

Practical tests have been made with a Norwegian ship equipped both with controllable pitch propeller and reversing gear on the engine. It took 50 per cent more time to stop the ship when using the propeller with fixed pitch than when the propeller pitch was varied.

One of the greatest advantages of a controllable pitch propeller for a submarine obtains when the ship is operating in submerged condition; this is due principally to the fact that the ship's resistance is increased five to six times the value which exists under surface condition. In the design of a submarine propeller, 85 to 90 per cent of the design considerations are devoted to surface operation, and therefore, it is evident that with the fixed propeller in a submerged condition the electric motor-propeller-ship combination will not be very advantageous.

However, by using a controllable pitch propeller the pitch can be adjusted to suit the increased resistance and available power in submerged condition as well as to give an ideal engine-propeller-ship combination in surface condition. With the same available battery capacity as when using a fixed propeller, the submarine may better utilize the electrical energy of the batteries whether it is desired to operate for the longest possible distance under the water, or to use the maximum possible speed for a short distance, either of which might be desired in an emergency. The submarine in submerged condition very often must execute extensive maneuvering, such as quick turns in small circles, coasting and sudden acceleration or deceleration of the ship. With a controllable pitch propeller these operations will be greatly speeded up and unparalelled flexibility will be obtained due to the complete utilization of the electric motor power, the simplification of the controls and the absence of the time lag caused by an intermediate person executing the officer's orders. Only two levers, one for each propeller, for controlling the ship's speed are needed.

Simplification of the Power Plant When a Controllable Pitch Propeller Is Used.

The performance and simplification of non-reversible Diesel engines for propelling ships are well known from the Diesel-electric drive. However, comparing features in any other system,

the striking simplicity and increased flexibility of the whole power plant when used in connection with a controllable pitch propeller should be pointed out. Diesel-electric drive in a ship has many advantages over the direct drive and although more flexible than the latter, a fixed propeller in such an installation cannot under all conditions utilize the available horsepower unless the electrical transmission is designed to function as a perfect torque convertor, which, for practical reasons, such as cost and size of the motor, is not usually done. The generators, motors, controls, wiring and the many instruments and switches are all sources of numerous troubles. The weight of the generators and motors is considerable and the cost of the electrical equipment, not to mention the high fuel consumption per useful horsepower due to the transmission losses, is appreciable.

The direct Diesel drive in a ship having fixed propellers, with the resulting inflexible power transmission to the propellers, has the disadvantage of complicated engines and, as a consequence, is less reliable and more expensive than the non-reversible engines. For instance, it is estimated that when designing a new 4-stroke cycle engine it will cost about 20 per cent more to include the reversible feature. Of course, this difference in price will be somewhat reduced when quantity production is reached. Besides the complicated reversing mechanism, there are the reversing valve arrangement for the service pumps, the intricate control station for the engines and the shaft brakes with their controls, etc.

Compare the Diesel electric drive and the direct Diesel drive in ships driven with fixed propellers with the possibilities offered when the pitch of the propellers can be varied. In the latter case, simple non-reversible engines, stripped of all the above-mentioned electrical and reversing equipment, are used. The engines run in one direction only and utilize the inertia energy of the whole rotating system when maneuvering. An immediate reaction to the required variation in engine power, due to the simple control of the propeller pitch, is obtained. A control is provided which, due to its simplicity can always be located in the navigation or maneuvering center of the ship.

Propeller Manufacturing.

In regard to the manufacturing difficulties it is believed that the controllable pitch propeller has a better chance to produce an equal and correct helix for all the blades. This is of great importance for good efficiency, especially for propellers of high speed. Warping of the blades is unavoidable with a completely cast and heat treated propeller, and it is then difficult to adjust the blades to exactly the same helix.

Dr. Nordstrom also points out that with controllable pitch propellers it is possible to correct incompleteness in the calculation. With such propellers we can, after the ship and the propellers are built, try out the most suitable pitch and RPM. in regard to horsepower and freedom from vibration. With fixed propellers these changes sometimes are very costly even if the ship has been model-tested in a test tank, because vibration difficulties are frequently the cause of propeller changes. It may be mentioned that such ships as *Queen Mary* and *Normandie* now have their third set of propellers installed.

The writer includes in this article two of Dr. Nordstrom's propeller diagrams which are believed to be illustrative of what can be gained in speed and thrust when using a controllable pitch propeller. They concern installations in a tow-boat and in a freighter, respectively. Referring to Figures 30 and 31,

D = Diameter of the propeller.
H = Pitch.
N_o = Rated shaft horsepower.
N = Value of horsepower less or equal to rated shaft horsepower.
n_o = Rated revolution of shaft.
n = Value of revolution less or equal to rated shaft revolution.
T = Propeller thrust.
η = Propeller efficiency.

From Figure 30 it will be seen that it is possible to obtain a gain of speed of 5.5 per cent at 25 meters per second head wind (56 miles per hour) and 12.5 per cent when towing with about ⅓ speed in still water and no wind, may be obtained. The pulling power at the same speed will increase 32 per cent; i. e., the tow-

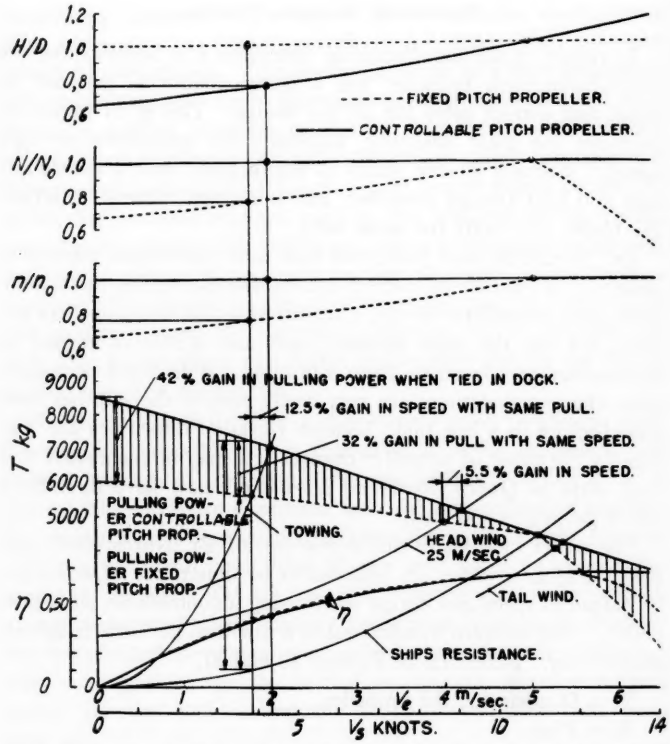


FIGURE 30.—DOCTOR NORDSTROMS' PROPELLER DIAGRAM FOR A TOW BOAT WITH CONTR'BLE AND FIXED PITCH PROPELLER. N = 450 HP., n = 132 RPM. SPEED IN STILL WATER AND NO WIND, 11 KNOTS, WAKE FACTOR = 0.10. PROPELLER DIAMETER = 2.86 METERS.

boat can, at the same speed, pull 32 per cent greater load. When pulling from dock an increase of 42 per cent pulling power will be obtained if the propeller pitch is suitably changed. In Figure 31 the curve chart represents a ship equipped with two engines connected by means of electromagnetic or fluid couplings and reduction gears to a single propeller shaft. It is found that when half the horsepower is used (one engine disconnected) the speed can be increased 9 per cent at full draft of the ship with no wind.

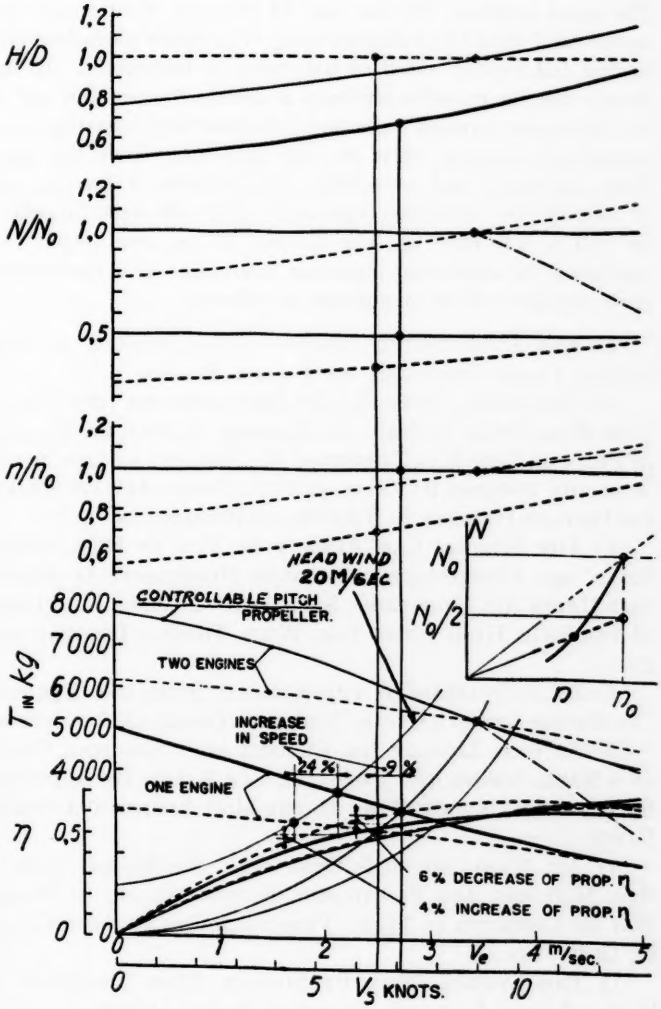


FIGURE 31.—DOCTOR NORDSTROMS' PROPELLER DIAGRAM FOR A FREIGHTER PROVIDED WITH TWO DIESEL ENGINES DRIVING ONE PROPELLER.

The speed increases not less than 24 per cent at 20 meters per second head wind (45 miles per hour) if propeller pitch decreases so that full rotative speed of the engine is maintained. In the former case the propeller efficiency is decerased 6 per cent and in the latter case increased 4 per cent. Another very intersting comparison can be made when the ship pulls away from the dock. Using one engine and controllable pitch propeller, it exerts a pull of 5000 kg., but when two engines are used with fixed propellers the pull is only 6200 kg., that is, only 24 per cent larger. In conclusion the eight most important advantages of a controllable pitch propeller will be summarized as follows:

(1) The Maneuvering Ability Is Far Superior to Any System Using Propellers with Fixed Blades.

(2) The Ship's Speed Can Be Regulated for Any Value from Full Speed to Zero. In Harbors, Canals, Locks, and in Fog, the Ship May Therefore Be Operated at Low Speed with the Engines Running at Full Speed, Always Ready for Instant Increase to Full Rated Horsepower.

(3) The Engines Can Always Be Run at Full Rated Speed, and Consequently Optimum Horsepower Is Available Under All Conditions, Such as No Load, or Full Load of the Ship, Head Wind, Tail Wind, Towing, Icebreaking, Etc.

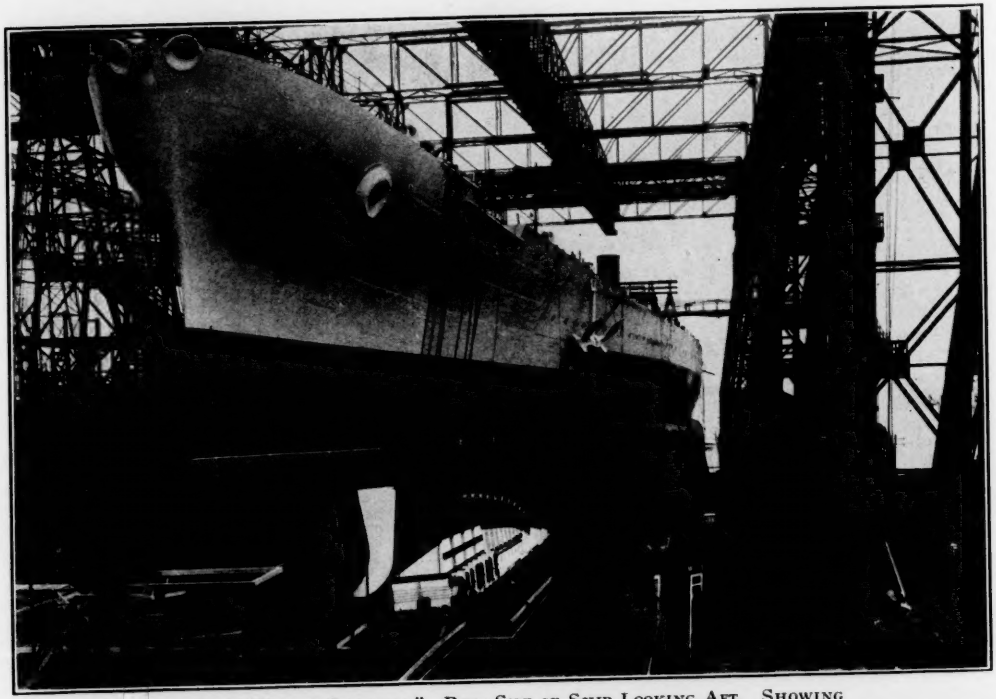
(4) By an Increase of the Rotative Speed the Engines Can Better Produce Excess Power for Overload Conditions.

(5) At Part Load of the Engines as at Cruising Speed of a Naval Vessel the Propeller Can Easily Be Adjusted So That the Engines Work in the Most Economical Speed Range.

(6) All Maneuvering Is Done from the Bridge, Therefore Mistakes Are Practically Eliminated or, if Made, May Be Corrected in Time. Personnel Can in Some Cases Be Decreased.

(7) The Engines Will Be Simpler, More Dependable, Have a Longer Life and, Therefore Lower Upkeep.

(8) Dangerous Torsional Vibration Can Easily Be Avoided.



U. S. S. "North Carolina"—Port Side of Ship Looking Aft. Showing Fore-Poppet, Sub-Poppet and Shoring.

THE LAUNCHING OF THE *NORTH CAROLINA.*

By Lieutenant Commander Leonard Kaplan, U. S. N.,
Member.

The author, in a technical manner, covers a type of event which has been absent from the American scene for many years—the launching of a battleship.

The U. S. S. *North Carolina* (BB55) was launched from Building Ways No. 1 at the New York Navy Yard, Thursday, 13 June, 1940, at 4:16 p. m.

It was a good launching—but so, too, are most launchings. And yet, notwithstanding the regularity with which vessels are launched, and the generally satisfactory manner in which these operations are carried out, the launch of a ship is always a matter of serious concern to her builders. Particularly is this true in the case of a large ship, where the launching operation takes on more the character of an individual problem. Here also the designer is handicapped by the paucity of performance data on undertakings of comparable size, and the lack of uniformity in launching practice among the several shipyards equipped to build the largest types of vessels.

The Launching Problem.

The launching problem is of especial interest because the forces involved are of extraordinary magnitude. In effect, a great mass, aggregating many thousands of tons, must be shifted from the foundation of blocks and shores on which it has been erected, to an inclined track (or groundways), and then allowed to slide down this track into the water. The operation is the more remarkable in that the actual transference and release of the huge mass are accomplished in the hours immediately preceding the launch.

U. S. S. "North Carolina"—Port Side of Ship Fore-Poppet, Sub-Poppet and Shoring.

The essential calculations include an investigation of the stability * of the vessel as it becomes water-borne, and an examination of the forces acting on the ship's structure during the run down the ways. The requirement for stability will be satisfied if the moment of buoyancy (displacement) about the end of the ways is always greater than the moment of weight about the same axis—that is,

$$Bd_1 > Wd_2$$

in the diagram of forces, Figure 1.

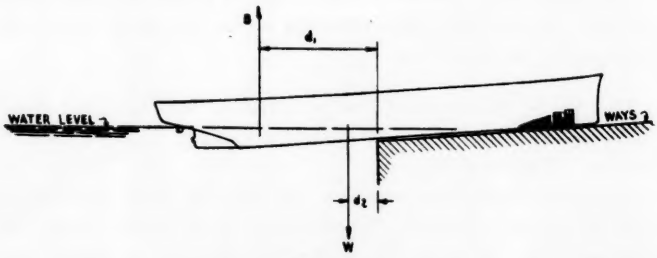


FIGURE 1.

Moment of displacement and moment of weight curves for the *North Carolina* are shown in the plot of launching curves, Figure 3. The amount by which the moment of displacement (or buoyancy) exceeds the moment of weight is called the "margin against tipping." It will be noted that there is such a margin, or anti-tipping moment, at each point of the ship's travel.

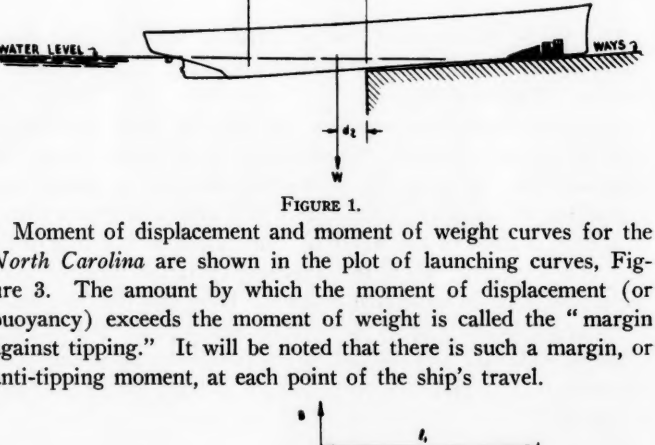


FIGURE 2.

* The ship must of course have transverse stability—be stable in the ordinary sense, that is—when fully water-borne.

The forces acting on the ship's structure during pivoting are of primary concern. Pivoting occurs when the moment of displacement about the forward end of the sliding ways is equal to the moment of weight about the same axis—that is, when

$$Bl_1 = Wl_2$$

in the diagram of forces, Figure 2.

There is sufficient buoyancy then to lift the stern and thus cause the ship to turn or "pivot" about the forward end of the launching cradle. The point in the ship's run where pivoting takes place is defined by the intersection (see Figure 3) of the curve of "moment of displacement" with the straight line which represents the constant "moment of weight." The pivoting load is the amount by which the weight of the vessel exceeds the displacement at this instant. It is the concentration of this very considerable load—3760 tons in the case of the *North Carolina*—under the forefoot of the vessel, that makes it necessary to build up the forward end of the launching cradle into a structure specially designed to insure an even distribution of pressure to the ship's hull. This structure is called the "fore-poppet."

Sizable loads are carried to the ship's structure also at the way ends, before pivoting occurs. The lessening force of reaction from the ground—itself the difference of the weight W and the buoyancy B—is applied over the diminishing area of contact between the sliding ways and the groundways. Calculation of way end pressures from the known force of reaction and area of application is necessarily predicated upon an assumed distribution of loading, and the values obtained are therefore approximate. Of lesser magnitude generally than the pivoting pressure under the fore-poppet, the pressure exerted by the way ends is frequently large enough and sufficiently localized to require extensive internal shoring of the ship's structure. The plot of way end pressures in Figure 3 shows a maximum for the *North Carolina* of 4.85 tons per square foot. This compares with a static or initial load of 2.31 tons per square foot, and a maximum pressure under the fore-poppet estimated at 6.55 tons per square foot.

To these basic considerations—a margin against tipping and means for carrying the loads caused by pivoting and way end

pressures—there is added the further requirement that the drop of the bow shall not be too great as the ship leaves the ways. Better still, if there is a sufficient depth of water at the outer end of the slip, the vessel will be fully water-borne when the fore-poppet passes over the end of the groundways, and no drop will occur. Instead, the ship will float clear, as was the case in the launch of the *North Carolina.* It will be noted from data tabulated elsewhere in this article that the distance slid to end of pivoting (float off) was 767 feet.

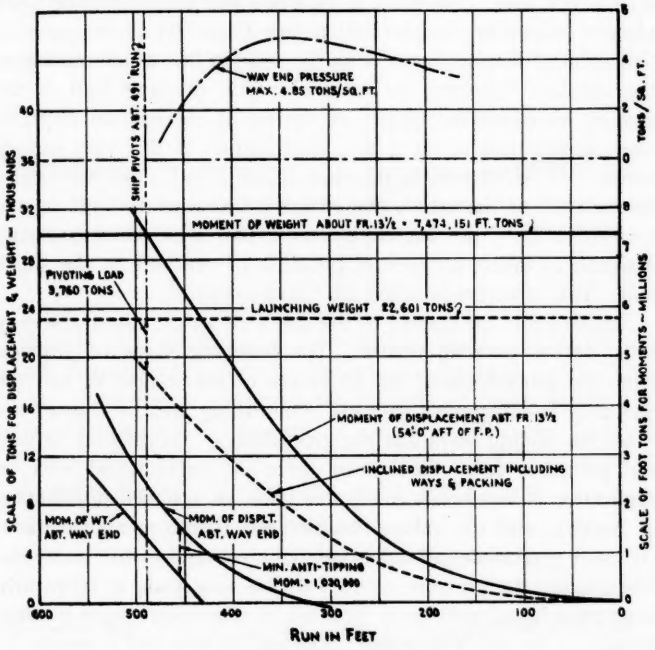

WAY END PRESSURE
MAX. 4.85 TONS/SQ.FT.
SHIP PIVOTS ABT. 491' RUN
MOMENT OF WEIGHT ABOUT FR. 13½ = 7,474,151 FT. TONS
PIVOTING LOAD
9,760 TONS
LAUNCHING WEIGHT 22,601 TONS
MOMENT OF DISPLACEMENT ABT. FR. 13½
(54'-0" AFT OF F.P.)
INCLINED DISPLACEMENT INCLUDING
WAYS & PACKING
MOM. OF WT.
ABT. WAY END
MOM. OF DISPLT.
ABT. WAY END
MIN. ANTI-TIPPING
MOM. = 1,030,000
SCALE OF TONS FOR DISPLACEMENT & WEIGHT – THOUSANDS
TONS/SQ. FT.
SCALE OF FOOT TONS FOR MOMENTS – MILLIONS
RUN IN FEET

FIGURE 3.—LAUNCHING CURVES.

ABSTRACT OF LAUNCHING DATA.

Weight of vessel, men and dunnage, tons	21,726
Weight of cradle, tons	875
Weight on grease, tons	22,601
Tons per square foot on grease (initial)	2.31
Tons per square foot on grease (maximum way end)	4.85

Tons per square foot on grease (maximum fore-poppet)		6.55
Maximum pivoting load, tons		3,760
Depth of water over way end (expected 12 feet–10⅝ inches), feet		13.48
Metacentric height (cradle included), feet		24.85
Distance slid to pivoting (time 32.5 seconds after starting), feet		491
Foot-tons moment against tipping		1,030,000
Distance slid to end of pivoting (float off), feet		767
Drop off at end of ways		None
Draft to bottom of keel with ways and packing under:		
Forward, feet and inches		10–2
Aft, feet and inches		23–8
Ground ways, cambered, slope under ship's center of gravity, inch per foot		½
Declivity of keel line, inch per foot		7/16
Keel line above grease at forward end of fore-poppet, feet		2.89
Number of ways		2
Length of sliding ways, feet		613
Width of sliding ways (each side):		
For 552 feet aft of forward end, feet and inches		8–0
For the remaining 61 feet, feet and inches		9–10
Effective area of sliding ways, square feet		9771
Triggers aft of forward end of ways, feet		466
Number of hydraulic triggers (there were 2 on each ways, all located in the same athwarthship line at Frame 126)		4
Sliding load on triggers before start, maximum (equivalent to 1682 pounds/square inch hydrostatic pressure), tons		924
Tumbling shores:		
(*a*) On center line skeg aft	6	
(*b*) Inboard of twin keels, in way of after poppet, 16 on each side, total	32	
(*c*) Outboard of launching ways, over middle body, 23 on each side, total	46	
	84	84
Dog shores		None used
Propellers		Not installed
Rudders		Not installed
Checking devices (the ways are pointed in the direction of the Williamsburgh Bridge, some 3700 feet away, the effect being to launch into the length of the East River)		None used
Speed during launch, maximum, feet/second		26.05
Launching lubricants:		
Base Coat:		
Name		Paragon Launching Stearine

Thickness, inch	½
Amount used, pounds	31,500

Slip Coat:

(*a*) For the first 552 feet on each side:

Name	Keystone Launching Grease
Thickness, inch	¼
Amount used, pounds	14,500

(*b*) For the last 61 feet on each side:

Name	Penola No. 4 and No. 5
Thickness, inch	¼
Amount used:	
Penola No. 4, pounds	1200
Penola No. 5, pounds	600

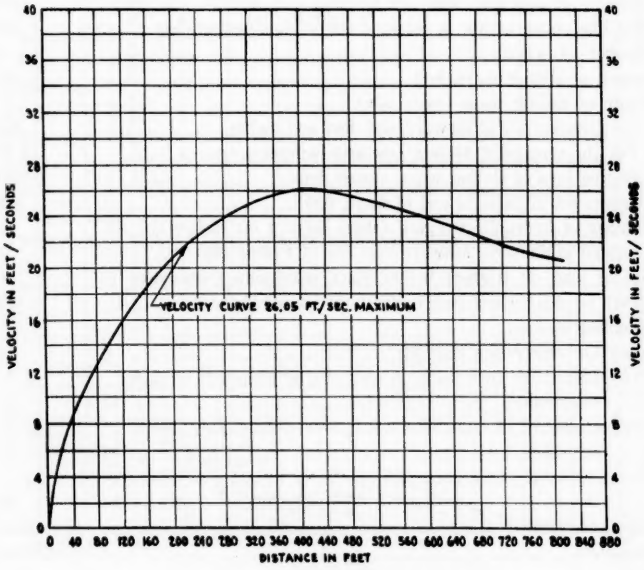


FIGURE 4.—SPEED-DISTANCE CURVE.

THE BUILDING SLIP.

Building Ways No. 1, in which the *North Carolina* was erected, is one of two adjacent shipways served by overhead bridge cranes. The cranes are carried on an elevated structure which spans both ways. The structure itself is carried on three rows of vertical

columns, the center row acting as a common support for the craneways of the two building slips. The distance between the rows of columns (158 feet 6 inches between centers) defines the limit of the slip transversely, while the length of the rows determines the length of the area that can be served by overhead bridge cranes.

The first of these limitations, that of width, is inherent in a building slip of this type, because the deadweight of structure rises sharply as the clear span of the bridges is increased. The effect of the restriction is to preclude the racking of material between the ship and the crane columns when the beam of the vessel is considerable, as in the case of the *North Carolina,* which has an extreme breadth of 108 feet 3 inches. This has of course no direct bearing on the launching problem.

The limitation of length, however, had a very definite effect both upon the construction and the launching of the vessel. At the time the ship was laid down, the overhead cranes had a run of 700 feet. Since the *North Carolina* measured 714 feet 0 inches between perpendiculars, the fore and aft position of the vessel in the slip was virtually fixed by the limitation of crane service. The area traversed by the cranes extended well into the flooded portion of the slip, because the runways, originally intended for the construction of a ship of more moderate length, had been carried out far enough for the cranes to plumb the water beyond the stern. The net result of building the vessel so far out into the river was to make it necessary to perform a great deal of work underwater in the preparation of the slip before construction was started—and again during the launching operation, as will be seen from the length of the launching schedule printed on another page.

The ways itself, the sloping platform upon which the ship was built, is essentially a 47-foot wide slab of concrete running down the middle of the slip. The 700-foot length and 27-foot difference in elevation between the two ends are divided into 29 approximately equal steps, so that the supporting surface is in effect a series of rectangular platens. The after 150 feet of the 700-foot long ways is below mean low water. A working platform of heavy wooden construction is built out on each side of the concrete ways to the full beam of the ship, and around the

stern at the after end of the slip. The platform serves the double purpose of providing better accessibility and reducing the length of the supporting shores.

The building ways is unique in that it has a permanent groundways over its entire length. The usual arrangement is to make the part of the groundways above water in portable sections, which are installed a few months before the vessel is ready to be launched. The portable groundways finds greater favor because it increases the working space under the ship, a very material consideration during the early part of the construction period.

Launching Arrangements.

The weight of the ship during the launching is carried by a cradle built to the shape of the hull and extending over almost the entire length of the vessel. The forward part of the cradle, which is of special construction designed to withstand the heavy pivoting pressure, is called the fore-poppet; and the after section, built up to accommodate the less regular contour of the stern, is called the after poppet. In the case of the *North Carolina,* the standard nomenclature has been elaborated to include the designation of "sub-poppet" for the section immediately aft of the fore-poppet. The arrangement of the launching cradle and poppets is shown diagrammatically in Figure 5 below.

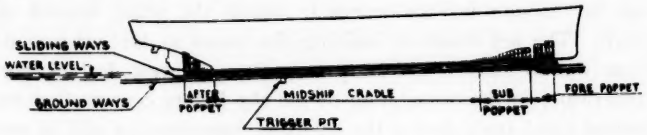


Figure 5.

The cradle rests upon a series of closely spaced wedges which separate it from the sliding ways underneath. The wedges are used to "ram up" the cradle to the hull on the day of launching, the effect of this being to take up any slack remaining between the fairing pieces and the shell, and to make the assembled cradle, wedges, and sliding ways sufficiently compact to function as a single supporting structure. Details of construction of the launching poppets for the *North Carolina* are shown in the ac-

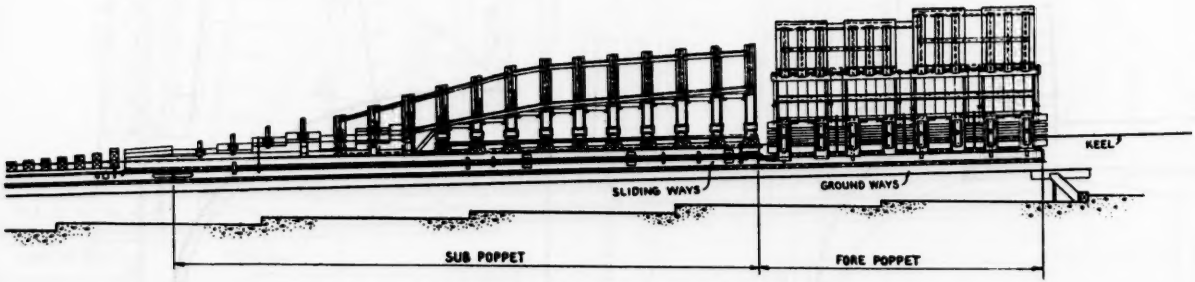


FIGURE 6.—PROFILE OF FORE-POPPET AND SUB-POPPET.

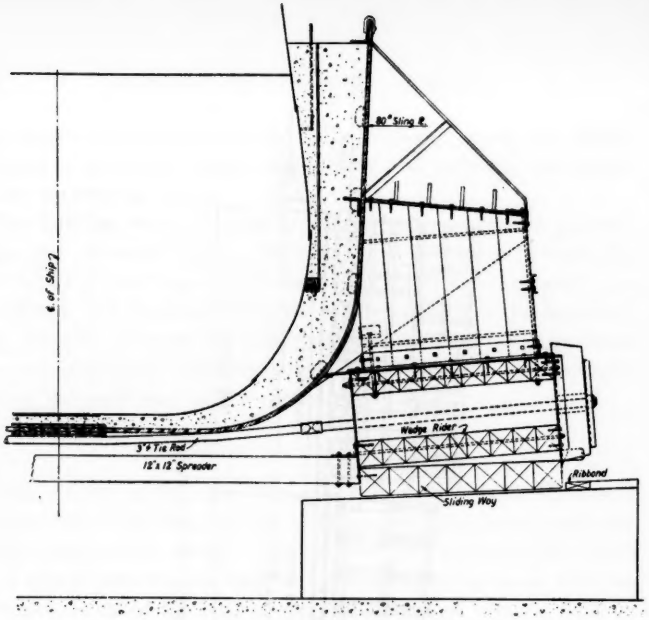


FIGURE 7.—TRANSVERSE SECTION THROUGH FORE-POPPET.

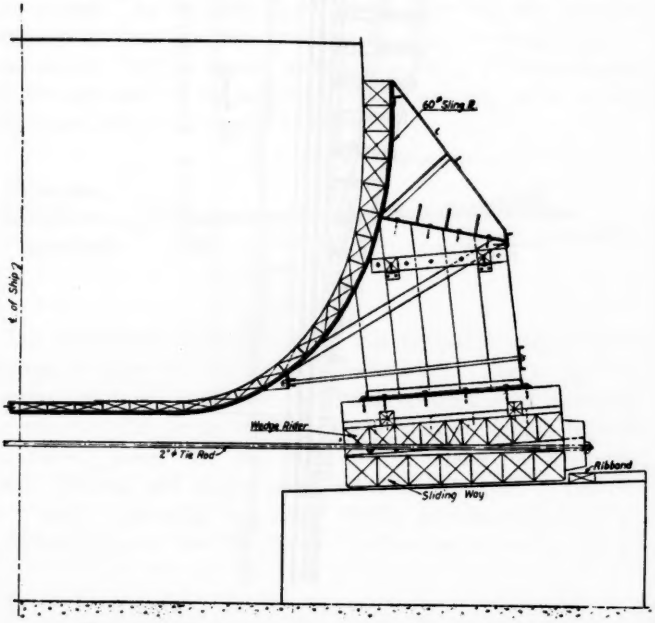


FIGURE 8.—TRANSVERSE SECTION THROUGH SUB-POPPET.

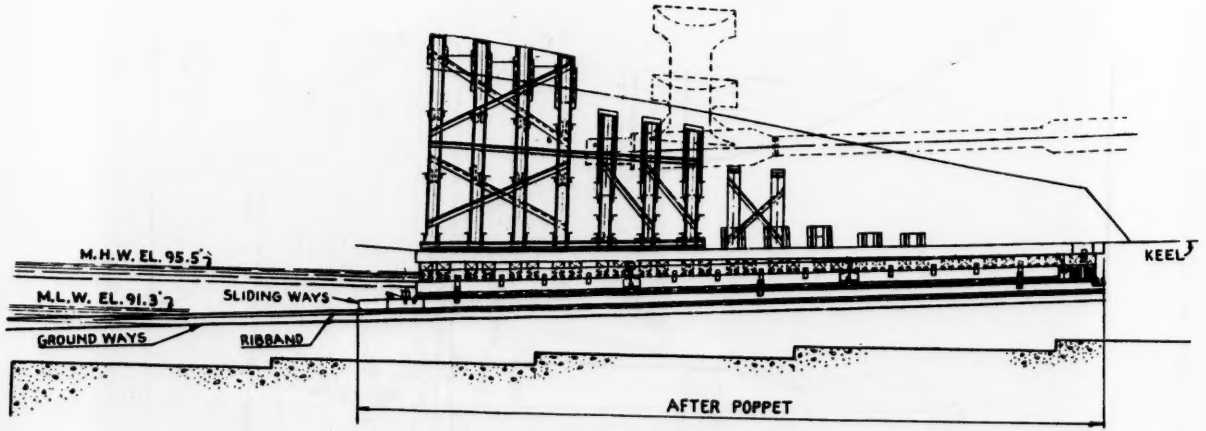


FIGURE 9.—PROFILE OF AFTER POPPET.

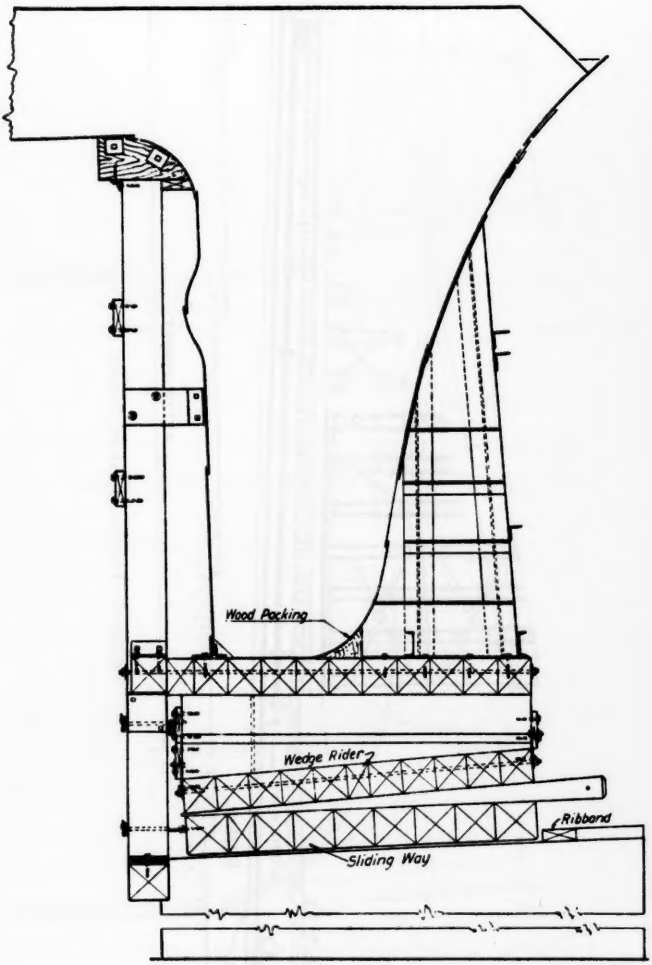
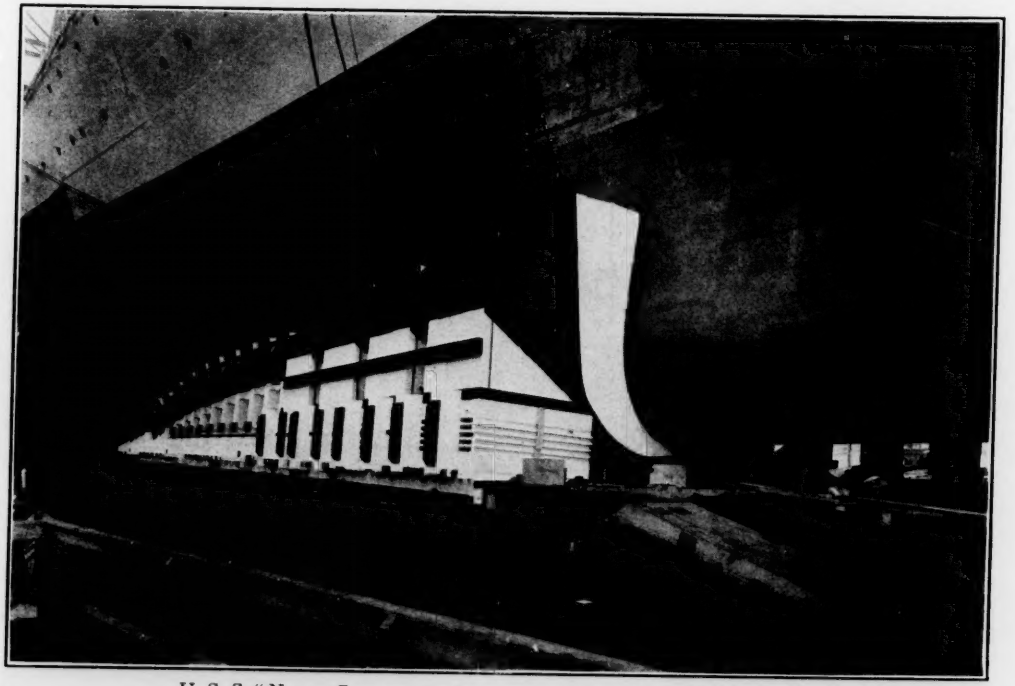


FIGURE 10.—TRANSVERSE SECTION THROUGH AFTER POPPET.



U. S. S. "North Carolina"—Starboard Side Looking Aft. Showing Backstop, Fore-Poppet, Sub-Poppet and Tricing Lines.

U. S. S. "North Carolina"—Launching—Sliding Down Way.

companying sketches and photographs. The sketches include sections through each of the three poppet structures, a profile of the assembled fore-poppet and sub-poppet, and a profile of the after poppet. An attempt has been made to give the photographs greater clarity by painting all the wood (and concrete) white, and the steel black.

The area of the sliding ways, and hence the width of the grease track, are fixed by the permissible loading of the launching lubricant. The initial (or static) pressure—the quotient, that is, of the weight of the ship and cradle divided by the area of contact with the groundways—is generally something less than 2.5 tons/square foot. Typical values for ships of comparable tonnage are given in the following table.

	Year Launched	*Total Launching Weight (Tons)*	*Initial Launching Pressure (Tons/Sq. Ft.)*
U. S. S. *West Virginia* (BB48)	1921	15,931	2.38
U. S. S. *Lexington* (CV2)	1925	26,897	2.45
U. S. S. *Yorktown* (CV5)	1936	16,249	2.54
S. S. *America*	1939	16,870	1.94
U. S. S. *North Carolina* (BB55)	1940	22,601	2.31

The lubricant consists of a base coat of hard grease (stearine), usually applied hot, and a slip coat of lighter grease put on by hand. The purpose of the base coat is to fair up any unevenness in the groundways, and to obtain a smooth, hard surface for the slip coat. The thicknesses of base coat and slip coat used under the *North Carolina* are about in line with current practice. These and other particulars regarding the lubricants are given in the abstract of launching data.

The weight of the ship rests on a foundation of keel blocks, shores, and cribs up until the commencement of the launching operation, which is essentially the task of transferring the weight from this foundation to the greased track of the launching ways. The deadweight of the cradle and sliding ways, up to this time, is carried on separators, commonly called "grease irons," which are inserted between the sliding ways and groundways, and have a thickness equal to that of the launching lubricant. The grease irons must be withdrawn and the cradle "rammed up" by

"rallies" on the wedges before the removal of blocks and shores can proceed. The latter are taken out progressively, working from aft to forward, the entire weight of the ship—except what is carried by the tumbling shores—resting on grease when the last blocks and shores have been cleared away from under the bow. The force acting down the ways, which is the component of gravity less the force of friction, increases as the weight of the ship is transferred to the grease track, and is resisted by launching triggers, usually located well aft in order to put the greater part of the sliding ways in compression. The ship is launched by releasing the triggers. The *North Carolina* had four, two on each ways, all located in the same athwartship line. They were hydraulically operated, and all piped into a common system so that the four triggers could be released simultaneously by opening one valve.

Schedules showing the order of events on the ground and aboard the ship, and the detailed procedure to be followed, are given below in the form in which they were issued before commencement of the launching operation. The scheduled times are for the most part in close agreement with those actually realized. The discrepancy in the final time of launching—the ship was launched at 4:16:00 instead of 4:11:30—is almost wholly accounted for by delay incurred in removing the last group of keel blocks, from Frame 54 forward. Prior to that time, little difficulty had been experienced in splitting out blocks within the scheduled periods.

SCHEDULED ORDER OF EVENTS ON THE GROUND.

First Period—Low Water Operations at Outboard (After) End of Shipways.

8:00 p.m., Wednesday, 12 June, 1940
to
4:00 a.m., Thursday, 13 June, 1940
Low Water—9:27 p.m. to 10:17 p.m.
High Water—3:12 a.m. to 4:02 a.m.

	Start	*Finish*
1. *Pull grease irons and remove dirt strips from under the two (2) after sections of the sliding ways, Nos. 10 and 9, port and starboard.*	8:00 p.m.	9:30 p.m.

NOTE: The removal of grease irons will begin at the after end of the sliding ways, the

	Start	*Finish*
crews on both sides working forward about in line with each other. The irons will be passed out from under the ship as quickly as they are drawn and carried to the racks located on each side of the trigger pit, where they will be hung in numbered positions.		
2. *Ram up wedges on two after sections, Nos. 10 and 9, of the sliding ways, port and starboard—3 rallies.* NOTE: The rallies will be of 5 minutes duration, with intervals of 3 minutes between.	9:30 p.m.	9:51 p.m.
3. *Remove thumb toggles on two after sections, Nos. 10 and 9, of the sliding ways, port and starboard.* NOTE: The thumb toggles will be carried to the grease iron racks as soon as they are removed, and stowed in numbered slots provided for that purpose.	10:00 p.m.	10:10 p.m.
4. *Split out keel blocks and remove "A" frames, starting from aft and working forward without interruption as far as, and to include, the last "A" frame supported block at Frame 147.*	10:00 p.m.	4:00 a.m.
5. *Remove the row of vertical shores on each side under the overhang of the stern, in approximate line with, aft of and between the rudderposts and the inboard strut castings.* NOTE: This will be done starting aft and working both sides forward together, so as to be in the same athwartship line, or very nearly so, with the men taking out the "A" frames under the keel.	10:00 p.m.	4:00 a.m.
6. *Remove the remaining (outer) shores from aft to Frame 155.* NOTE: These shores and the ones in item 5 above will be painted with a double red band, to indicate that they are to be removed during the first period of the launching operation.	10:00 p.m.	11:00 p.m.

	Start	*Finish*
NOTE: Removal of these shores will also be in step with the "A" frames, the effect being to take out keel blocks and adjacent shores at the same time.		
7. *Remove the working platform and the supporting (transverse) timber trusses as far forward as Frame 155.*	10:30 p.m.	4:00 a.m.
NOTE: This will be done as rapidly as the removal of the shores (item 6) from the working platform to the ship will permit.		
8. *Remove the two (2) rows of double red band shores on each side of the keel track, between the keel blocks and the launching ways, extending from Frame 146 to Frame 130.*	1:30 a.m.	3:00 a.m.
NOTE: These shores will be taken out immediately after removal of the outer shores (item 6) to Frame 155 has been completed. The same procedure of working from aft to forward will be followed, the crews working the four rows of shores together in approximate line athwartships.		
9. *Remove the six (6) vertical shores which alternate with the six (6) main tumbling shores under the center line skeg, and extend from Frame 146 to Frame 133.*	1:30 a.m.	3:00 a.m.
NOTE: Removal of these shores will be from aft to forward and in step with the 4 inner rows of shores (item 8) above.		
NOTE: The total number of double red band shores, including the 6 in this item, is estimated at 84.		
10. *Remove covering boards from exposed portion of grease track beyond the ends of the sliding ways.*	2:00 a.m.	4:00 a.m.
Second Period—Operations Preliminary to Main Launching Sequence.		
6:00 a.m. to 11:30 a.m., Thursday, 13 June, 1940.		
Low Water—10:00 a.m. to 10:50 a.m.		
11. *Pull grease irons and remove dirt strips from under the eight (8) remaining sections of the*	6:00 a.m.	8:00 a.m.

	Start	*Finish*
sliding ways, Nos. 8 to 1, inclusive, port and starboard.		
NOTE: These will be hung in numbered positions on racks provided for that purpose, on each side of the ship in way of the trigger pit, as explained in item 1 above. A final check will be made at this time to insure that all are in place and accounted for.		
12. *Remove all of the working platform (and the supporting timbers) from Frame 155 to Frame 136 that can be taken away without disturbing the shores that remain in until the Third Period of the Launching Operation.*	6:00 a.m.	8:00 a.m.
13. *Remove all single red band shores.*	6:30 a.m.	10:30 a.m.
NOTE: The single red band will serve to denote shores that are to be taken out during the Second Period of the Launching Operation. This group is made up for the most part of shores which will have to be split out. They include:		
(*a*) The 2 rows of short shores closest to and outboard of the launching ways, port and starboard, extending from Frame 146½ to Frame 45½.		
(*b*) One row of shores on each side, spaced two frames apart, between the keel blocks and the launching ways, extending from Frame 125 to Frame 38.		
(*c*) A single row of center line shores on the half frames, sandwiched in between the keel blocks in way of the sub-poppet, from Frame 29½ to Frame 17½.		
The shores in this group number about 262.		
NOTE: The plan of removal will be to begin aft and work forward about in line with—but not ahead of—the crews pulling the grease irons.		
14. *Saw through the supporting transverse members, and split out the inboard row of blocks*	6:30 a.m.	10:30 a.m.

		Start	*Finish*
	in the four (4) forward cribs on each side of the ship. Take out only the center inboard blocking of the 5th (aftermost) cribs.		
	NOTE: Partial demolition of the cribs at this time will serve to expedite their removal in the Third Period of the Launching Operation.		
15.	*Tow barrier out of slip.*	10:00 a.m.	10:15 a.m.
16.	*Ram up wedges on the eight (8) remaining sections of the sliding ways, Nos. 8 to 1, inclusive, port and starboard—3 rallies.*	11:00 a.m.	11:21 a.m.
	NOTE: The rallies will be of 5 minutes duration, with intervals of 3 minutes between.		
17.	*Remove thumb toggles on the eight (8) remaining sections of the sliding ways, Nos. 8 to 1, inclusive, port and starboard.*	11:21 a.m.	11:30 a.m.
	NOTE: The thumb toggles will be carried to the grease iron racks as soon as they are removed, and stowed in numbered slots. A final check will be made at this time to insure that all are in place and accounted for.		
	Third Period—Main Launching Operation.		
	12:00 noon to 4:11:30 p.m., Thursday, 13 June, 1940.		
	High Water—4:06 p.m. to 4:56 p.m.		
18.	*Split out keel blocks, remove shores and cribbing from aft to Frame 125.*	12:00 noon	12:35 p.m.
	NOTE: The procedure here will be to start aft and work forward, taking out keel blocks and adjacent shores at the same time. When cribs are encountered, removal of shores will be halted until the cribs have been split out.		
	NOTE: There are approximately 121 keel blocks and 544 shores—in addition to the 10 cribs—to be removed during the Third Period of the Launching Operation.		
19.	*Check Launching Triggers. Put on 200 pounds initial pressure.*	12:30 p.m.	12:40 p.m.
20.	*Complete removal of working platform and supporting structure from Frame 155 to Frame 136.*	12:30 p.m.	2:00 p.m.

		Start	*Finish*
	NOTE: This will complete the removal of the working platform for purposes of the launch. The planks and the timber trusses underneath will be torn out as rapidly as the removal of the shores (item 18) will allow.		
21.	*Split out keel blocks, remove shores and cribbing from Frame 125 to Frame 114.*	12:35 p.m.	1:10 p.m.
22.	*Split out keel blocks, remove shores and cribbing from Frame 114 to Frame 102.*	1:10 p.m.	1:40 p.m.
23.	*Check operation of jacks.*	1:30 p.m.	1:40 p.m.
	NOTE: A pressure of not more than 10 pounds may be applied to insure that the pistons of the four 300-ton jacks are free to move, and that they bear uniformly on the jack beams at the forward ends of the sliding ways.		
24.	*Split out keel blocks, remove shores and cribbing from Frame 102 to Frame 92.*	1:40 p.m.	2:10 p.m.
25.	*Split out keel blocks, remove shores and cribbing from Frame 92 to Frame 80.*	2:10 p.m.	2:40 p.m.
26.	*Split out keel blocks, remove shores and cribbing from Frame 80 to Frame 68.*	2:40 p.m.	3:10 p.m.
27.	*Split out keel blocks, remove shores and cribbing from Frame 68 to Frame 54.*	3:10 p.m.	3:40 p.m.
28.	*Captain of the Yard report river clear*	3:30 p.m.	
29.	*Split out keel blocks, remove shores and cribbing from Frame 54 to bow.*	3:40 p.m.	4:08 p.m.
30	*Remove gangway*	3:45 p.m.	3:50 p.m.
	NOTE: There will be only one gangway to take in—a brow from the crane structure to the main deck at Frame 55 on the starboard side.		
31.	*Display red flag to river from outboard end of center catwalk in overhead crane structure of No. 1 Building Ways.*	4:00 p.m.	
32.	*Officer-in-Charge aboard ship report readiness for launch.*	4:05 p.m.	

	Start	*Finish*
33. *WHEN DIRECTED BY THE LAUNCHING OFFICER, knock out shores from under and drop locking bars (safety latches) of the four (4) launching triggers.*	4:09 p.m.	
34. *All hands clear of ship.*	4:10 p.m.	
35. *Notify sponsor's stand.*	4:10 p.m.	
36. *Sound warning whistle—one blast.*	4:11 p.m.	
37. *LAUNCH.*	4:11:30 p.m.	

SCHEDULED ORDER OF EVENTS ABOARD SHIP.

1. *Sound all tanks.*	10:00 a.m.	10:30 a.m.
NOTE: This will serve as a preliminary run to insure that all hands understand what they are required to do. Any difficulties encountered are to be taken up by the supervisor with the Watertight Integrity Officer, and ironed out at this time. Mimeographed forms have been provided, which list the tanks to be sounded. Individual sheets will be issued to each man to record his own soundings. The sheets will be collected and delivered by messenger to the Watertight Integrity Officer as quickly as possible.		
2. *Check operation of sound powered telephones.*	1:00 p.m.	1:15 p.m.
NOTE: Outlets will be provided at the bow and stern, and at the control station amidships. An additional line will be run from the control station to the 200 Kw. Diesel generator set in No. 2 machinery space.		
3. *Muster on station men detailed to make observations and records for scientific section.*	1:30 p.m.	1:45 p.m.
NOTE: These men will take readings on crushing indicators, observe pivoting of the ship, and record such other data as required by the Design Superintendent.		
4. *Check readiness of connections for receiving steam and water from Penobscot immediately after launching.*	2:00 p.m.	2:15 p.m.
NOTE: The *Penobscot* will come alongside on the starboard side at about Frame 56,		

	Start	*Finish*
where temporary connections to steam lines and fire mains have been provided.		
5. *Make watertight closures (trial run).*	2:30 p.m.	2:40 p.m.
NOTE: These are the remaining closures, left open until the day of launching, required to carry out the plan of watertight integrity. There are only five closures to be made, all hatches, in locations as listed below:		
Frames *Off C.L. to Deck*		
Hatch to Passage B-403-L.. 81½–83½ 18′–0″ S 3rd		
Hatch to Compt. B-2 ..... 89– 90 9′–11½″ P 3rd		
Hatch to Compt. B-3 ..... 102–104 9′–11½″ S 3rd		
Hatch to Compt. B-4 ..... 113–115 9′–11½″ P 3rd		
Hatch to Compt. B-5 ..... 123–124 9′–11½″ S 3rd		
6. *Disconnect air lines.*	2:30 p.m.	3:00 p.m.
7. *Reopen watertight closures (item 5).*	2:45 p.m.	2:55 p.m.
8. *Sound all tanks.*	3:00 p.m.	3:20 p.m.
NOTE: This will be the final check before launching.		
9. *Muster on station men detailed to handle lines.*	3:00 p.m.	3:30 p.m.
10. *Muster on station men detailed to emergency (portable) pumps.*	3:30 p.m.	3:45 p.m.
NOTE: The portable pumps will be located in the following stations:		
(a) *No. 2 turret on second platform deck.* One large simplex pump for use with either air or steam.		
(b) *No. 3 turret on second platform deck.* One large simplex pump for use with either air or steam.		
(c) *Steering gear enclosure (Compartment C-424-E).* One single-acting reciprocating pump for use with either air or steam.		

	Start	Finish
(d) *Frames 10-12—Second platform deck.* One single-acting reciprocating pump for use with either air or steam.		
(e) *Frames 10-12—Second deck.* One single-acting reciprocating pump, for use with either air or steam.		
(f) *Frames 130-132—Main deck, forward of No. 3 turret.* One single-acting pump will be held on deck in readiness for emergency use at any location to be designated. The supervisor-in-charge will take up his station just before launching with the crew detailed to man this pump.		
11. *Muster on station men detailed to check sea valves and operate Engineering (fixed) pumps.*	3:30 p.m.	3:45 p.m.
NOTE: This will be the final check, before launching, of sea valves still accessible for inspection, and supervisors-in-charge will report readiness for their respective parts of the ship at this time.		
NOTE: The two pumps manned by this (X38) detail are:		
(*a*) The fire and flushing pump located in the evaporator room.		
(*b*) The fire and bilge pump in No. 4 machinery space.		
12. *Start 200 Kw. Diesel generator set in No. 2 machinery space (preparatory run) and secure after 15 minutes' running.*	3:30 p.m.	3:45 p.m.
13. *Disconnect electric light and power leads from shore.*	3:45 p.m.	3:55 p.m.
14. *Anchor detail at stations ready for letting go.*	3:45 p.m.	3:50 p.m.
15. *Make watertight closures (final).*	3:45 p.m.	3:55 p.m.
16. *Break telephone connection to ground.*	4:00 p.m.	4:02 p.m.
17 *Start 200 Kw. Diesel generator set and take temporary light load.*	4:06 p.m.	

		Start	*Finish*
18.	**SCHEDULED TIME OF LAUNCHING.*	4:06 p.m.	
19.	*Reopen watertight closures.*	4:10 p.m.	4:20 p.m.
20.	*Signal tugs to come alongside and proceed to Pier G.*	4:10 p.m.	4:30 p.m.
	NOTE: Twelve tugs will be available to take the ship in tow. The vessel will be berthed at Pier G, port side to, the fore-poppet about in line with the center of the 350-ton hammer head crane. Floats or lighters will be used to breast the stern out from the dock enough to throw the bow in as much as possible.		
21.	*Sound all tanks.*	4:15 p.m.	4:35 p.m.
	NOTE: This will be the first check after launching.		
22.	*Inspect sea valves.*	4:30 p.m.	5:00 p.m.
	NOTE: This will be a water-borne check of the sea valves referred to in item 11 above—those that are not in sealed compartments.		
23.	*Sound all tanks.*	6:00 p.m.	6:20 p.m.
	NOTE: This will be the second and final check after launching.		

The working force employed on the day of launching is necessarily large, because the tasks to be performed are essentially man-power operations. The ground crew for the *North Carolina* numbered 655. Of this total, the 359 shipwrights accounted for the largest single group. They carried out all the work within the immediate vicinity of the ship, which amounted to the major part of the launching procedure. The riggers and laborers removed the blocks and shores from the building ways as rapidly as they were made available by the shipwrights, and assisted generally in pulling grease irons and handling shores. There were 238 in the ship detail, which included a large working party of Yard riggers and enlisted men to handle lines; plumbers to sound tanks and stand by portable pumps; and a special detail of machinists and

* The launching was scheduled for 4:06:00 p.m. at the time this order was issued. The subsequent change to 4:11:30 p.m. was considered too minor to warrant republication of the schedule.

electricians to run the ship's 200 Kw. emergency Diesel generator set in No. 2 Machinery Space. Recapitulations by trades of the personnel of the ground crew and the ship detail follow.

RECAPITULATION OF PERSONNEL ON THE GROUND.

Officers	10
Civilian Personnel.	
Masters and Foremen	25
Shop. No.	
X-02—Transportation	16
X-11-12-26—Structural trades	40
X-51—Electricians	47
X-56—Plumbers and Pipefitters	17
X-61—Shipwrights	359
X-71—Painters	16
X-72—Riggers and Laborers	77
X-79—Yard Fire Department	5
Draftsmen	24
Photographers	9
Miscellaneous	10
Total	655

RECAPITULATION OF PERSONNEL ON THE SHIP.

Officers	5
Enlisted Men	100
Civilian Personnel.	
Shop No.	
X-11-12-26—Structural trades	21
X-38—Machinists (Outside)	26
X-51—Electricians	8
X-53-56—Coppersmiths and Pipefitters	43
X-61—Shipwrights	4
X-72—Riggers and Laborers	17
Draftsmen	13
Miscellaneous	1
Total	238

U. S. S. "North Carolina"—Launching—Sliding Down Way.

The task of removing the poppets and the cradle after the vessel is launched comes as something of an anti-climax, and must be provided for in the original design. It is not an easy problem when the ship is large, and the weight of the material to be removed necessarily considerable. The usual procedure is to drop the poppets in the water alongside the pier, and to snake out the cradle and sliding ways in sections afterwards, if they have not already floated clear of the vessel. A variation of this procedure is to use a flooded drydock for the same purpose, the advantage in this being that the poppet structures can be broken up into smaller parts after the dock has been pumped down, and salvaged more readily. A different plan was followed in the case of the *North Carolina.* The ship was berthed under the new 350-ton hammerhead crane, where the fore-poppet and the lower part of the after poppet were lifted clear immediately after launching. The fore-poppet weighed 230 tons, and the portion of the after poppet removed, 110 tons. The vessels was towed to No. 4 Drydock the following morning, where it was docked with what remained of the launching cradle, on blocks built up 20 inches more than their normal height above the floor of the dock. This was accomplished without casualty, the packing and wedges, and the sections of sliding ways falling clear of the rows of blocking as the dock was pumped down. Since the vessel was to remain in drydock for something over two months, ample time was available to break up and remove the launching material from the dock in lifts of convenient size.

In conclusion, it seems well to point out that there are no hard and fast rules for launching a ship. The procedure followed in most cases will reflect the evolution of ideas or the experience of a particular building activity. There is little inclination to experiment because of the risk involved—a tendency rather to hold to previous practice in almost too exacting detail. And yet, it is up to the innovator to prove his case. The substitution of new methods will be warranted only if it can be shown that definite economies will be effected without increase of launching hazard.

NOTES.

INDEX TO NOTES.

THE UNITED STATES LINER "AMERICA."
—Shipbuilding and Shipping Record, London, England, May 23, 1940.

RADIOGRAPHIC DEVELOPMENT IN THE CASTING INDUSTRIES.
—Metals and Alloys, East Stroudsburg, Pa., January, 1940.

THE SECOND GROUP OF NATIONAL DEFENSE FEATURE TANKERS.
—Pacific Marine Review, San Francisco, Calif., April, 1940.

HIGH PRESSURES AND TEMPERATURES PLANNED FOR MARINE FIELD.
—Combustion, New York, N. Y., April, 1940.

SUPERPOSED TURBINE BLADE RESEARCH.
—Mechanical Engineering, Easton, Pa., April, 1940.

LOAD TESTS OF A COMBUSTION GAS TURBINE.
—Brown Boveri Review, Baden, Switzerland, April, 1940.

MISCELLANEA OF INTEREST TO NAVAL ENGINEERS.

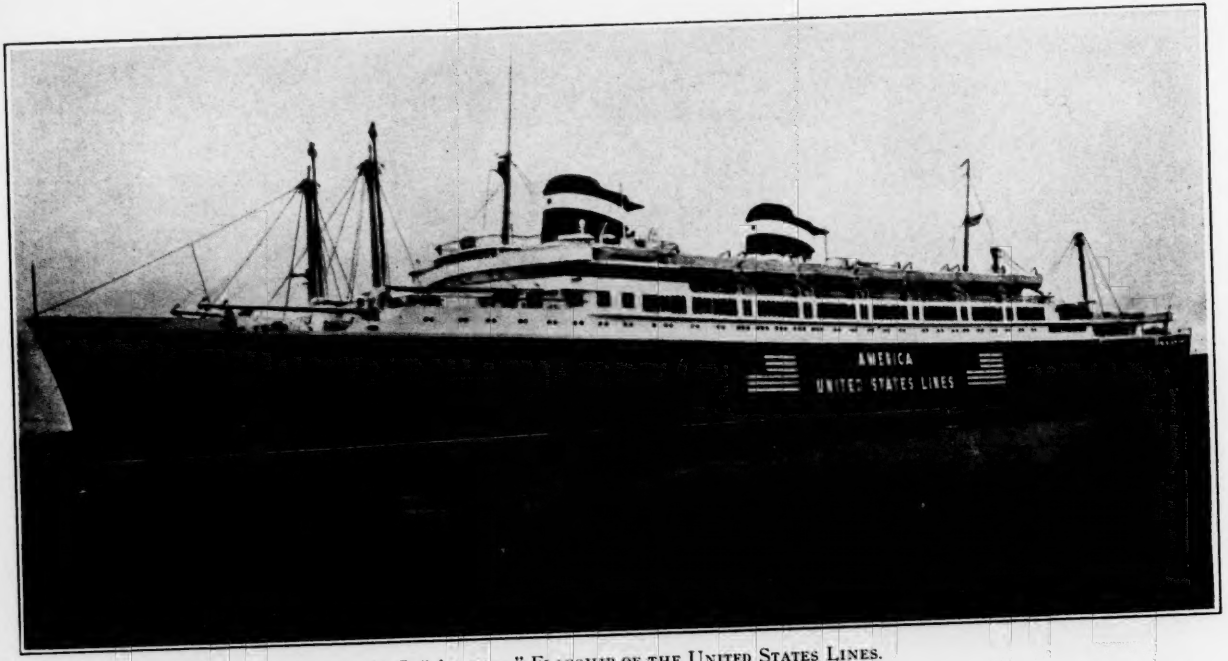
Speed and Power of Destroyers.
—Shipbuilding and Shipping Record, London, England, Feb. 1, 1940.

Forty Years' Advance in Efficiencies.
—Engineering, London, England, April 22, 1940.

Resuscitation from Shocks.
—Transactions of the Institute of Marine Engineers, London, England, March, 1940.

Engine-Room Ventilation in Liners.
—The British Motorship, London, England, June, 1940.

A Study of Fuel Specifications.
—The British Motorship, London, England, June, 1940.



S. S. "America," Flagship of the United States Lines.

THE UNITED STATES LINER *AMERICA*.

The following is an abstract of a paper presented by Harold F. Norton and John F. Nichols at the Spring Meeting, held at Newport News, Virginia, of the Society of Naval Architects and Marine Engineers. Mr. Norton is a Naval Architect of the Newport News Shipbuilding and Dry Dock Company and Mr. Nichols is Chief Engineer of the same firm. Reprinted from the May 23, 1940, issue of Shipbuilding and Shipping Record, published at 33 Tothill Street, Westminster, London. The *America* completed her first sea trials on June 5, 1940. The photograph of the S. S. *America* is reprinted from the Marine Journal, published at 5 Beekman Street, New York City.

The design for the S. S. *America* finally selected by the owners, the United States Lines Company, and approved by the Maritime Commission and the Navy, was developed by Gibbs & Cox, Inc., naval architects and the owners' representatives. It was the culmination of a long series of designs developed by Gibbs & Cox and also, at the owners' request, by the Newport News Shipbuilding & Dry Dock Company. At the various stages of development the owners had pointed out the features they considered undesirable and selected those they considered desirable in each design. They had also consulted the Maritime Commission and the Navy. The result is an arrangement which represents what seems the best composite of the ideas of all, even though it may not completely meet the ideals of each.

The work had begun well before 1935 and was not finally completed until several months after the signing of the contract on October 21, 1937. During that time eight different tentative designs had been produced by the Newport News Shipbuilding & Dry Dock Company and at least an equal number by Gibbs & Cox. The designing started with vague ideas of a modernized duplication of the *Manhattan* and *Washington* and ended with a vessel distinctly different in general arrangements and slightly larger in all principal dimensions.

SEPARATED BOILER ROOMS.

Perhaps the most unusual feature of the Gibbs & Cox arrangement finally adopted by the owners is the separated boiler rooms, with engine room between, the uptakes going up through the machinery hatch. All three main machinery spaces are protected from collision damage by deep fuel-oil tanks on each side. It is our understanding that this arrangement for the vessel was favored by the Navy as desirable in case of use as a Naval auxiliary and by the owners as interfering least with public space arrangements.

CHRONOLOGY OF CONSTRUCTION.

Contract signed	October 21, 1937
Lines approved and dimensions settled	February 8, 1938
Keel laid	August 22, 1938
Launching	August 31, 1939
Sea trial scheduled for	About June 15, 1940
Delivery scheduled for	About June 28, 1940

At the laying of the keel, Admiral Land, Chairman of the Maritime Commission, drove the first rivet. Mrs. Roosevelt, First Lady of the United States of America, acted as sponsor at the time of the launching.

The technical features of the launching are described in such thorough detail in Mr. Vincent's article in the October, 1939, issue of "Marine Engineering and Shipping Review," that it is unnecessary to repeat them here, except to mention that the launching weight was 16,870 tons, the pressure on the grease 1.94 tons per square foot and the inclination of the straight ground ways 9/16 inch per foot. The vessel started promptly when the launching triggers were released and the maximum sliding speed was 18.4 feet per second.

The *America* was designed as a flagship for the United States Line fleet for a fast and efficient passenger and freight service with the *Washington* and *Manhattan* between New York and Hamburg, calling at Cobh, Plymouth, Southampton and Havre.

In developing the lines as finally approved, about fifty models were tried in the Newport News model tank. In all, over 5000 test runs were made. These models were not all from the same set of dimensions, as the length, beam and draught of the ship varied during the development of the design. Also, bulbous bows, blister sections amidships, fine *versus* full waterlines and many other variations of form were tried.

Experiments were made with a corresponding 20-foot model in the Washington Model Basin to determine the best trace for the bilge keels, the best shape for the bossings and the best propellers, in order that the ship as a whole should be as efficient as possible hydrodynamically. The curves of power and speed for the final design are shown in the graph above.

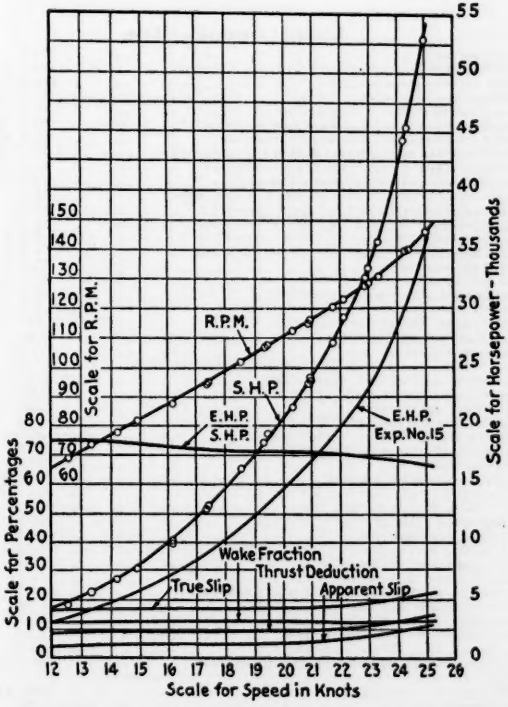
The *America* is built and equipped according to the regulations of the Public Health Service, the Fire Underwriters, the British Board of Trade, the Bureau of Marine Inspection and Navigation and the American Bureau of Shipping, under whose special survey the vessel is given the highest rating A-1-E for North Atlantic passenger service.

In appearance the *America* is distinctive. The slightly curved raking stem, the high freeboard forward and the streamlined treatment of the bridge front and superstructure, together with the unusual design of the wide-spaced stacks with their sampan tops, to deflect smoke away from the sports deck, present an appearance that makes the *America* readily distinguishable from all other existing vessels.

The general structural design of the hull is as shown on the midship section and on the profile and arrangement plans.

There are ten decks, the sports, sun, promenade, upper, main, "A," "B," "C," "D" and "E" decks, respectively. "E" deck is fitted only in No. 1 hold. "D" and "C" decks extend forward and aft of the machinery spaces. The "B," "A," main and upper decks are continuous for the length of the ship. The promenade deck extends for a length of 513 feet 9 inches and is the main strength deck to the hull for the midship portion of the vessel. It has on each side an athwartship overhang of 2 feet beyond the line of the hull immediately below. The tumblehome to the underside of the promenade deck is 3 feet 7½ inches. The sun deck is above the promenade deck and has a length of 360 feet 8 inches. The sports deck is above the sun deck and has a length of 343 feet 9 inches. The hand-ball court screens and two streamlined windbreaks on each side help to make this deck usable for deck games under all but very bad weather conditions. As the stack is fitted with Vortex soot collectors, as well as a special sampan top, it is expected that this deck will be kept quite clear of smoke and cinders.

No expansion joints are fitted in the superstructure, the deck plating being increased in thickness as necessary to function as a part of the main hull girder system. The resulting stresses are considered to be conservative and within good practice limits for medium steel. No special steels are used in the vessel.



Data for speed-power curves estimated from self-propelled tests at U.S. Experimental Model Basin with model No. 3525 using propeller No. 1803–04. Ship data: Length, 689 ft.; beam, 92 ft.; displacement, 34,732 tons; trim, even keel; wetted surface, 81,410 sq. ft.; appendages—rudder, bilge keels, thick bossings. Number of propellers, 2; diameter, 19·5 ft.; pitch at 2/3 radius, 19,458 ft.; number of blades, 4; mean width ratio, 0·24; ratio of projected area to disc area, 0·417; blade thickness fractions, 0·0535; direction of rotation, outboard.

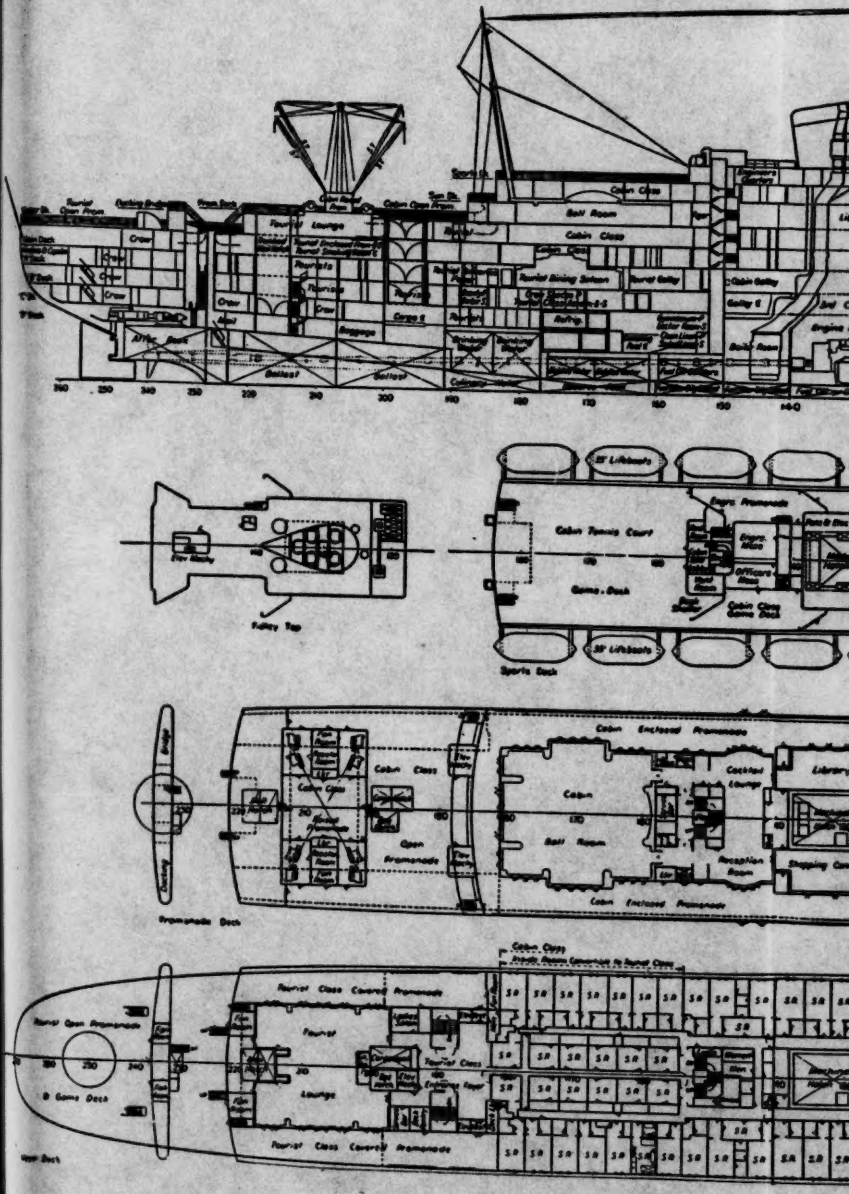
SPEED AND POWER CURVES.

Principal Dimensions and Data

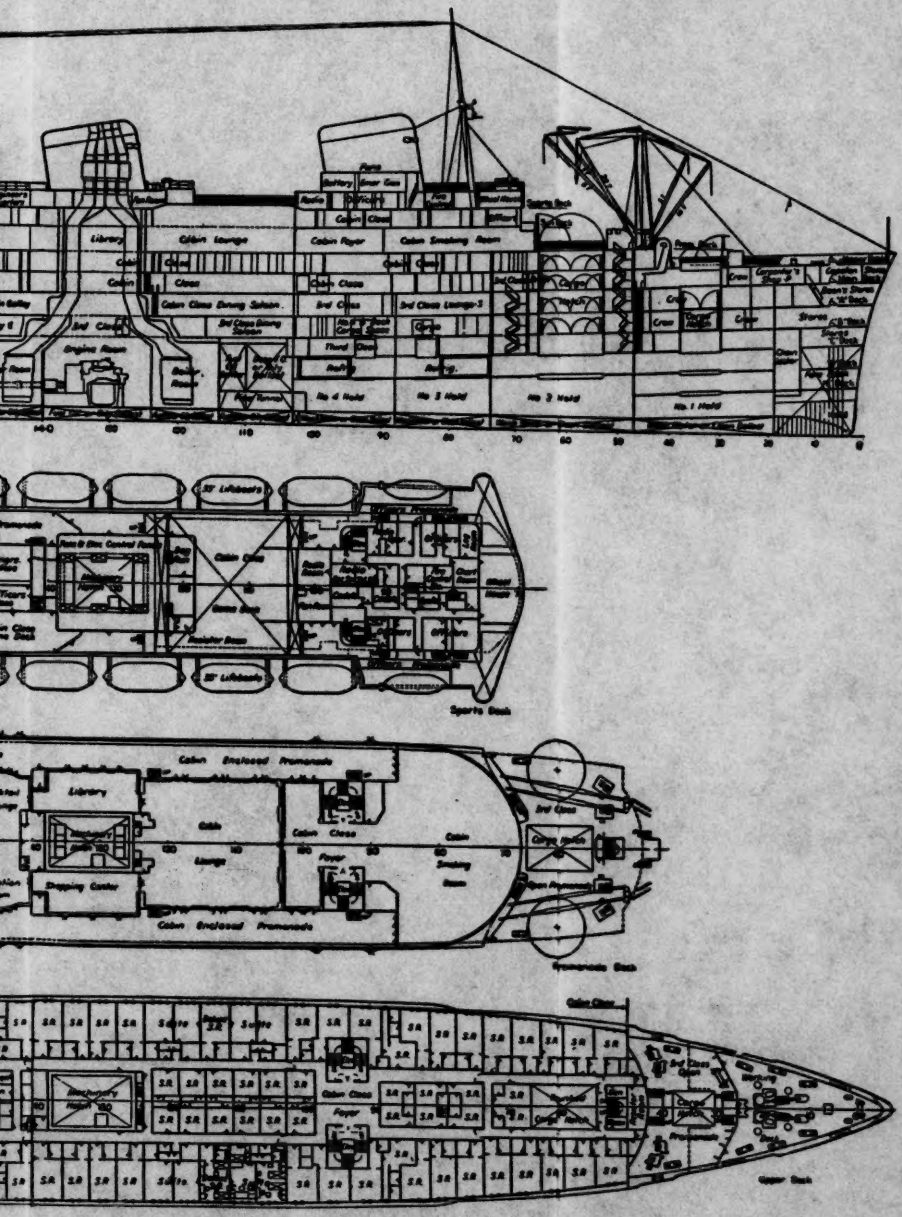
Length overall	723 ft.
Length on 32-ft. 6-in. waterline	690 ft. 3 in.
Length between perpendiculars	660 ft. 6⅛ in.
Beam moulded, maximum	93 ft. 3 in.
Beam moulded at 32-ft. 6-in. waterline	91 ft 11½ in.
*Depth moulded to sports deck at side amidships	92 ft. 4½ in.
*Depth moulded to sun deck at side amidships	82 ft. 4½ in.
*Depth moulded to promenade deck at side amidships	73 ft. 4½ in.
*Depth moulded to upper deck at side amidships	64 ft. 5½ in.
*Depth moulded to main deck at side amidships	55 ft. 5½ in.
*Depth moulded to main deck at lowest point of sheer	55 ft.
Height between main and "A" decks, frame 162	10 ft.
Height between "A" and "B" decks, frame 162	9 ft.
Height between "B" and "C" decks, frame 162	8 ft. 6 in.
Height between "C" and "D" decks, frame 162	9 ft. 6 in.
Load draft, moulded	32 ft. 6 in.
Load draft to bottom of keel	32 ft. 8⅝ in.
Sheer forward at forward perpendicular	11 ft.
Sheer aft at after perpendicular	2 ft. 10 in.
Camber, upper deck and above, in 86 ft.	3 in.
Camber on main deck and below	None
Displacement, full load, tons	35,440
Tons per inch immersion at 32-ft. 6-in. moulded draft	110
Deadweight, full load, tons, approximate	13,061
Gross tonnage, approximate	27,000
Net tonnage, approximate	15,000
Block coefficient at full load draft	0·5859
Maximum section coefficient at full load draft	0·9772
Prismatic coefficient at full load draft	0·5996
Waterplane coefficient	0·7147
Cargo, general, cubic feet, bale	about 265,000
Cargo, refrigerated, cubic feet, net	about 33,500
Ship's cold storage, cubic feet, net	about 34,350
Baggage space, cubic feet, net	about 19,650
Mail space, cubic feet, net	about 30,000
Shaft horsepower, normal	34,000
Shaft horsepower, maximum	37,400
Designed speed, in excess of	22 knots
Cruising radius, miles, at 22 knots at load draft, about	11,000
Cruising radius as above at 15 knots	18,000
Fuel capacity, 97 per cent. full, tons at 37·1 cu. ft.	4,938
Tanks available for water ballast, tons	2,238
Fresh water, tons, 100 per cent. full	4,733
Compartmentation, flooding and stability flooded	3 compartment
Metacentric height estimated to be necessary for 3-compartment flooding (without margin)	about 4 ft.†
Cabin passengers	543
Tourist passengers	418
Third-class passengers	241
Total passengers	1,202
Crew	643

* Note.—Lowest point of sheer is aft of amidships, frame 162.

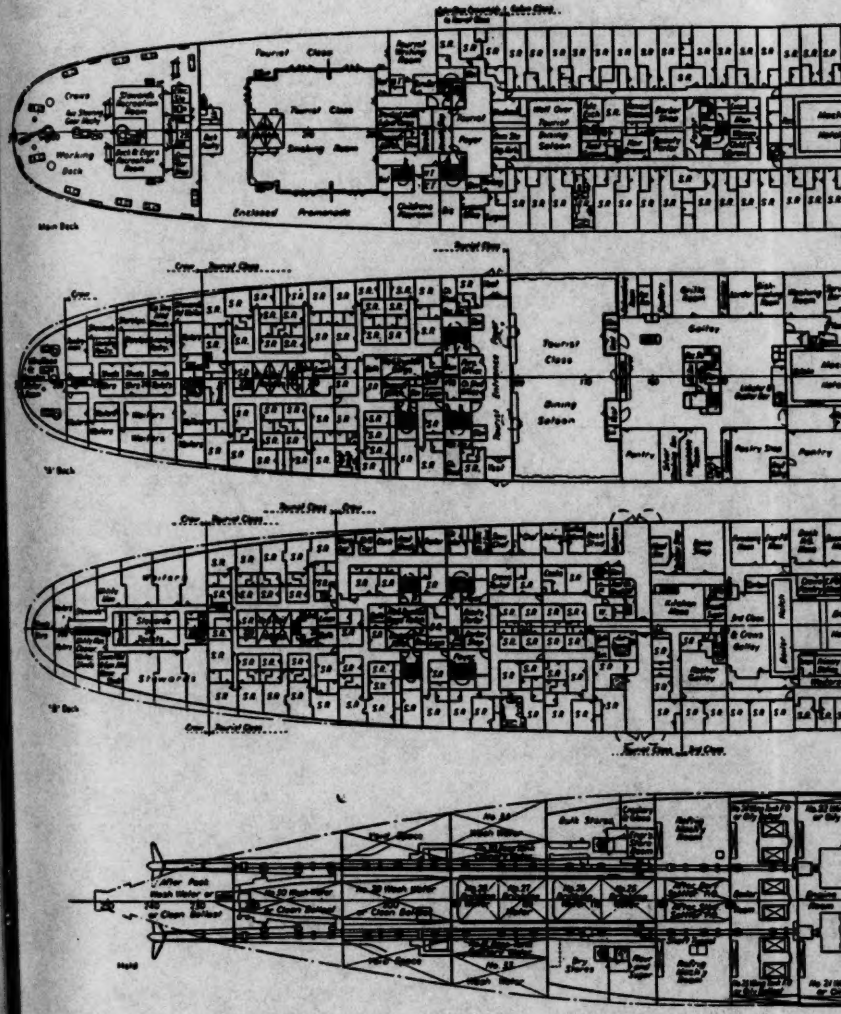
† The inclining experiment has not been performed at the date of writing. This GM plus a margin is anticipated. The estimated necessary GM varies between about 3·90 ft. and 4·2 ft. depending upon the operating condition.



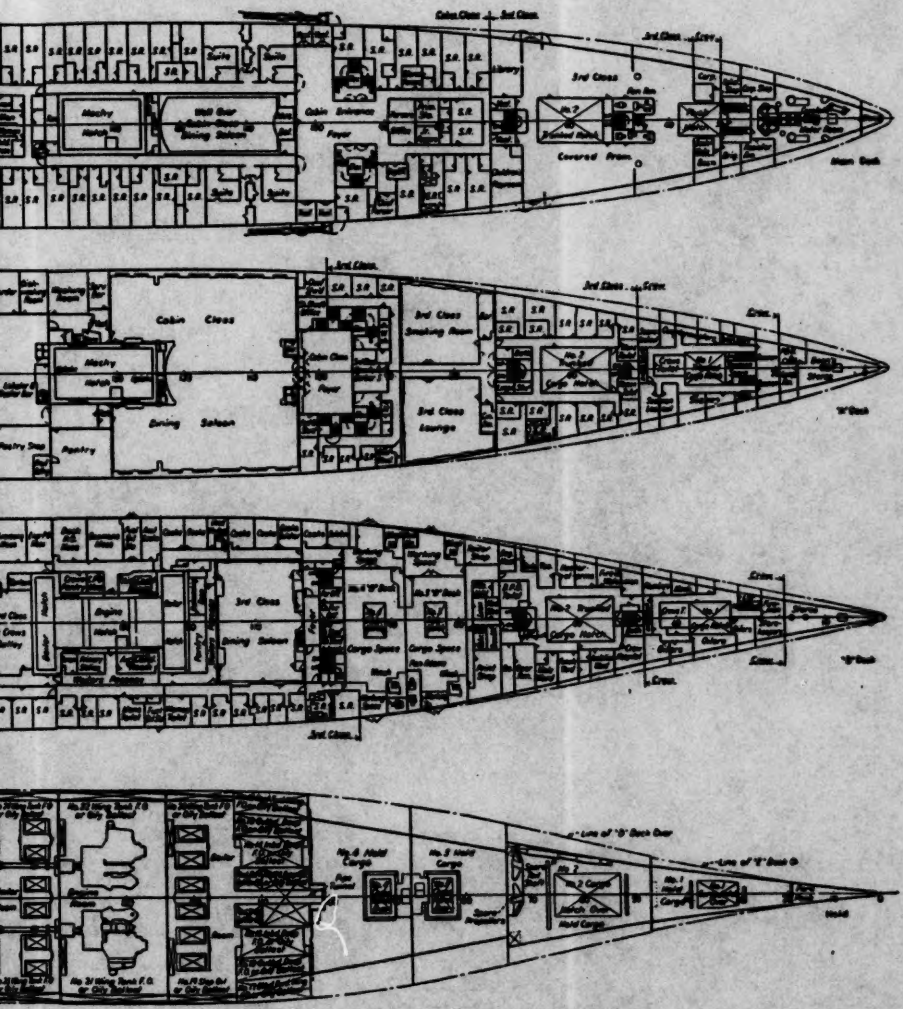
General Arrangement of the Pass



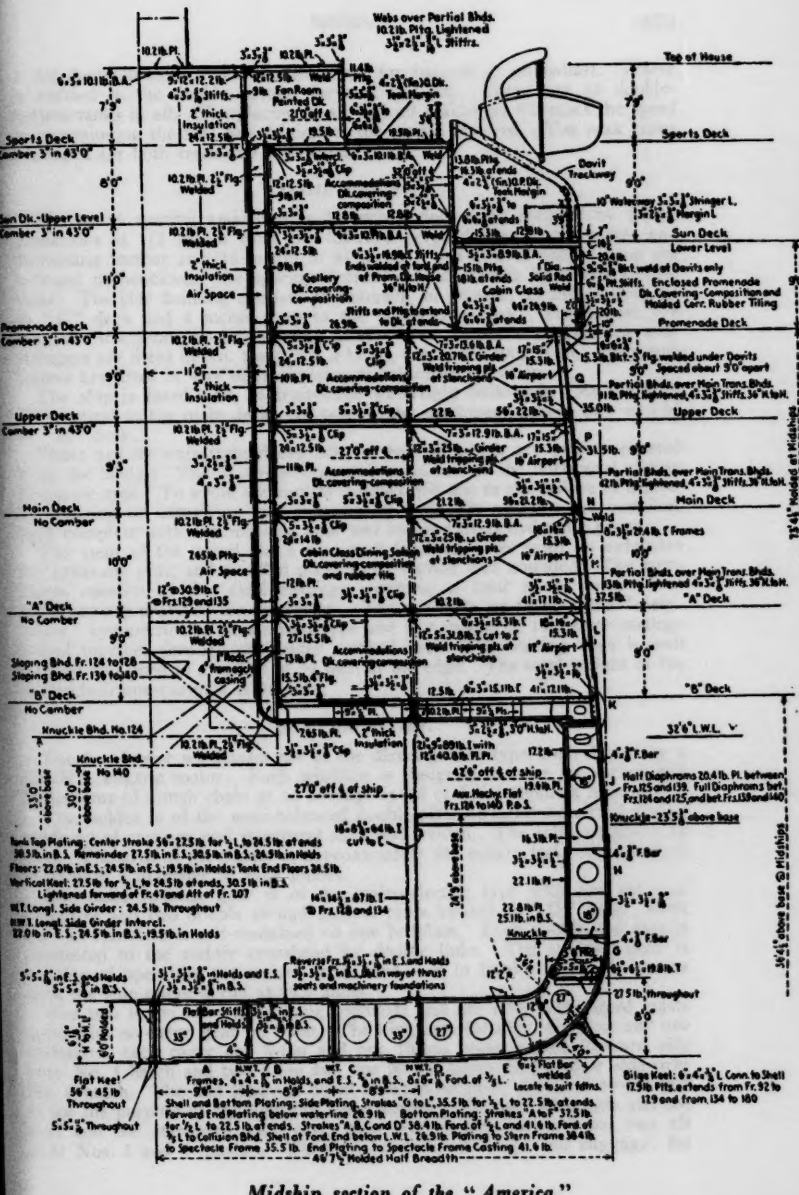
THE PASSENGER STEAMSHIP "AMERICA."



General arrangement of the passenger steamship "America," built and engined
"America," which is the largest vessel to be built in the U.S.A., has a gross t



d engined by the Newport News Shipbuilding and Dry Dock Co., Newport News. The a gross tonnage of 27,000 tons and is propelled by steam turbines totalling 37,400 S.H.P.



Midship section of the "America"

All double-bottom tanks amidships are for fuel oil or oil ballast. Water is carried in the double bottom forward and aft. There are 35 double-bottom tanks in all. The placing of all fuel oil amidships increases the speed of provisioning the ship and minimizes the piping required. The peak tanks are piped for both fresh and salt water.

FRAME SPACING.

The frame spacing amidships is 36 inches, decreasing forward by 1-inch gradations at 1/5 length from the forward perpendicular to 27 inches and decreasing further at 1/15 length to 24 inches. Aft at 9/10 length from the forward perpendicular the spacing decreases to 24 inches by 2-inch gradations. The side framing consists of channels, in general, 10 inches deep up to "C" deck and 8 inches deep to the promenade deck. The frames are joggled throughout, no liners being fitted. Web frames in conjunction with stringers are fitted in No. 1 and No. 2 holds, due to the long span. Deep web frames are fitted in way of the bossing to minimize vibration.

The ship is fitted with 14 transverse watertight bulkheads. At the ends they extend to the main deck. In way of the machinery spaces they extend to "A" deck.

There are 59 watertight doors, all but two of which are power-operated from the bridge. The 57 power-operated doors are of the horizontal sliding hydraulic type. To avoid a complete disablement due to the failure of a pipe, by collision or otherwise, two independent hydraulic systems are installed, each complete with accumulator tank and motor-driven hydraulic pump.

The stem of the vessel has a cast-steel forefoot, fitted with an extension for paravane gear, and a rolled-plate upper section well rounded. The stern frame consists of five steel castings weighing a total of 41 tons. Two gudgeons are provided for the rudder. The spectacle frame is of the "clear-water" symmetrically-streamlined type and is composd of two steel castings bolted together through the center vertical keel. The trailing edge is well tapered and fined to a radius of ⅛ inch at the edge. The total weight of the spectacle-frame castings is 65 tons.

DECK MACHINERY.

The two bower windlasses are of the direct-geared type, each driven by a 100-HP. electric motor. Each windlass is designed to raise its anchor and 30 fathoms of 3-inch chain at an average speed of five fathoms a minute.

The rudder is of the semi-balanced double-plate streamlined airfoil design made up of castings and structural steel and welded. The total weight is 33 tons. The area of the rudder is approximately 425 square feet. The stock is solid with a diameter of 24½ inches.

The main steering gear is of the hydro-electric type with two pairs of cylinders and two double 18-inch hollow rams located on "D" deck. Each pair of cylinders is self-contained on one bedplate. Each double end ram is connected to the rudder crosshead by double links. The steering gear is capable of operating the rudder from hard-over to hard-over in 30 seconds with the ship going ahead at 23.5 knots.

The two masts have telescoping topmasts and are radio and signal masts only, with no cargo gear on them. The kingposts forward, one port and one starboard, each carry one 5-ton and one 10-ton boom on the forward side over No. 1 hatch and two 5-ton and one 20-ton boom on the after side over No. 2 hatch. The fact that one boom forward on each side is of 10-ton size is for the purpose of handling the spare anchor. The two kingposts aft, one port and one starboard, each carry two 5-ton booms forward and two aft over Nos. 5 and 6 hatches. The first hatch is for cargo and baggage; the

second is the mail hatch, which has a partition in the middle to aid in segregating the mail for different ports of call. Nos. 3 and 4 hatches each have two 3-ton jib cranes and cargo ports on "B" deck. Each cargo port has a hinged platform which may be rigged outside to assist in receiving lightered cargo. The ship's refrigerated and other stores are loaded through stowing ports on "C" deck.

There are 20 cargo winches, driven by 35-HP. motors. Four of the winches are located below decks for serving cargo cranes. The remaining 16 winches are on the weather deck for serving the cargo booms.

The refrigerated-cargo spaces forward on "D" deck are all air cooled and arranged for temperatures as low as 10 degrees F. Each space is insulated with Johns-Manville incombustible BX-4 material between frames or beams and covered with ⅛-inch steel plating.

The ship's cold-storage spaces on "C" and "D" decks aft are all brine cooled, arranged for temperatures from minus 10 degrees to plus 45 degrees F. and are insulated and lined in the same way. The entire construction is incombustible.

MACHINERY.

The design of the machinery, as well as of the hull, represents a compromise between the ideas of at least four elements, the United States Maritime Commission, the owners, the U. S. Navy, and the builders. The machinery layout was naturally strongly affected by the fact that the vessel was primarily to be a running mate for the *Manhattan* and *Washington*, which had already been in service for several years.

A careful preliminary study was made of the service record of the *Manhattan* and *Washington* with a view to retaining what was good and making such improvements as seemed practicable. For this reason many of the general features were retained, twin screws, triple turbines with reduction gears, six boilers. The steam pressure and temperature have been advanced somewhat from the previous practice and the high-pressure turbine was made of the impulse type throughout, increased in speed and fitted with double-reduction gearing. The intermediate and low-pressure turbines are of moderate speed and single reduction is retained, the main gear being the largest marine gear produced in this country. Due to the increased beam it was found practicable to arrange all the propulsion auxiliaries in the engine and boiler rooms, thus reducing the length of the machinery space and of the main steam pipes. The main condensers were located beneath the low-pressure turbines instead of above, thereby insuring reliable drainage from the turbines at all times. Condenser circulation by scoops was retained. This arrangement, which is an off-shoot from Naval practice, has been found most satisfactory in service, is probably of equal efficiency with pumps and has the advantage of saving space and of eliminating the necessity of maintaining two vital constant running auxiliaries.

RADIOGRAPHIC DEVELOPMENT IN THE CASTING INDUSTRIES.

This article by Charles W. Briggs, Technical Advisor, Steel Founders' Society of America, reflects the modern trend of progressive thought in the castings industry. Reprinted from January, 1940, issue of Metals and Alloys, published by Reingold Publishing Corp., East Stroudsburg, Pa.

In 1895, Roentgen, in an experiment involving the emission of ultra-violet light from high vacuum discharges, discovered the X-ray. However, it was not until 1912 that much industrial significance was attached to the idea of non-destructive testing of materials by X-rays. It was during that year that Coolidge introduced a new type of tube which could be operated continuously at higher voltages, used greater currents, and thus increased both the penetrative power and the intensity of the rays. The advent of the Coolidge tube marked the beginning of the effective use of radiography in industry. However, records of the use of X-rays in the casting industries are not as early as this. Some little experimental work was carried on in connection with the non-destructive testing of aluminum castings during the World War, but nothing in the way of routine testing.

EARLY HISTORY.

From 1915 to 1920, radiographic facilities and technique improved greatly with the introduction of better power plants, tubes of longer life, and more sensitive film.

It has been only since 1922 that radiography has had serious use in the metal industries. It was during that year that the U. S. Army Arsenal at Watertown, Mass, installed a 200,000-volt, 5-milliamp X-ray equipment. This installation raised the thickness limit of steel which could be radiographed commercially from 1½ to 3 inches. The arsenal staff, under the direction of Dr. H. H. Lester immediately undertook to inspect all types of castings that were being made for the U. S. Army.

In 1923, Dr. Lester published an account of an examination of steel castings by radiography. His investigations allowed him to make statements concerning the application of the X-ray method. He found that, for practical working conditions, detectability may be placed at 2 per cent, which means that a discontinuity of 0.05 inch in diameter can be detected in steel 2.5 inches thick.

The results that were being obtained at Watertown created considerable interest among engineers generally. Mechanical engineers were much interested in the commercial application of non-destructive testing to pressure equipment castings, and during 1925 Moultrop and Norris published an article on the X-ray examination of steel castings in "Mechanical Engineering." The work described in the paper was carried out in testing steel castings that were subject to a steam pressure of 1200 pounds. They pointed out that radiography was sufficiently sensitive to show in a steel section a discontinuity of 2 per cent, and that flaws equivalent to 10 or 20 per cent or more of the section were only of importance if their detection and estimation were possible with a satisfactory degree of accuracy.

Shortly after this date, engineers began to demand an X-ray inspection of important castings for high-pressure steam plants. Light aluminum alloy castings began to be prominently used in all types of structural application which required radiography.

The non-destructive testing of metals took another step forward when, in 1929, Dr. R. F. Mehl, with the aid of Drs. Barrett and Doan, completed a systematic research which established the use of gamma rays from radium for the inspection of metal. The gamma-ray method of radiography was immediately applied to steel casting inspection by the U. S. Navy. Important hull and propulsion castings were studied, which resulted in a widespread use of gamma-ray radiography during the next eight years throughout all the Navy Yards and Naval inspection centers.

FIRST PORTABLE X-RAY SETS.

The first portable X-ray sets were introduced in 1930, and in 1932 a 300,000-volt Coolidge tube became available, increasing the thickness of steel that can be commercially X-rayed from 3 to 4½ inches. Since then higher voltage tubes have been built, until now one foundry organization is equipped with a 400,000-volt X-ray unit capable of making diagnostic radiographs through 5 inches of steel.

Today, the non-destructive testing of castings, while still considered a special test, is not looked upon by the foundryman with awe and uncertainty. The foundryman has found it to his advantage to keep up with the developments being made in radiography. To help him keep abreast of radiographic activities, a committee on the radiography of castings was organized by the American Foundrymen's Association in 1935. It was their duty to act as a fact-finding committee and to submit a yearly report concerning the progress being made in radiography as applied to the manufacturing, control and inspection of castings. This the committee has done.

The radiographic committee of the A. F. A. held an informative session in 1936 for the purpose of discussing the appearance of radiographs and the interpretation of the shadow markings. The manner in which radiography may be used in the foundry as an aid in solving production problems was also studied. The cost of radiographic equipment was reviewed, as well as presentation of the tabulated results of a questionnaire on the subject of radiography in the steel foundry, which received industry-wide distribution.

SYMPOSIUM ON RADIOGRAPHY.

In 1936, another important radiographic event took place which caused considerable interest throughout the metal industries. The American Society for Testing Materials held a Symposium on Radiography and X-Ray Diffraction. From the excellent information presented at this symposium the foundry industry learned that the X-ray method and the gamma-ray method were not competitive methods of non-destructive testing, but instead, were complementary in scope. Foundrymen learned that the advantages of X-ray radiography were those of short exposure times and pronounced contrast of image, while the advantages of gamma-ray radiography were portability and the degree of latitude (the extent of variation in metal section that may be satisfactorily radiographed at one time).

Furthermore, foundrymen learned that if they were required to radiograph light castings on a production basis, they should resort to the use of X-ray equipment, while if they were interested in the non-destructive testing of castings from 4 to 10 inches thick, they may consider using gamma rays.

To carry on the good work that had been started so well by the symposium, the A. S. T. M. organized, in 1937, Committee E-7 on Radiographic Testing, with the hope that all advancements in this country in the technique of radiographic testing and the formation and adoption of radiographic standards would come to the attention of the Committee.

The foundry industry is represented on Committee E-7 and particularly on Sub-Committee I, which concerns itself with the Radiography of Cast Metals. The work of the Sub-Committee is in the formative stage, yet it has prepared a revision of the Recommended Practice for Radiographic Testing of Metal Castings (E 15-29), and has proposed for adoption as a tentative standard, the Gamma-Ray Radiographic Standards for steel castings for steam pressure service, as prepared by the Bureau of Engineering, Navy Department.

The foundry industry is not opposed to the establishment of sets of radiographic standards that are fair in their treatment of the subject. In fact, they welcome them, providing the standards are prepared for castings of a specific application, such as the above Navy standards which are limited to steel castings intended for steam pressure service. The reason that standards of this type are willingly accepted by foundrymen is that they set up standards which may be observed and used as guides. This is quite different from some vague ideas on rejection limits that repose in the minds of the purchasers of castings or their authorized inspectors. The presence of standards sharply defines the situation.

RADIOGRAPHIC INTERPRETATION.

One of the most serious drawbacks in the adoption of radiographic methods by casting producers has been the fear of radiographic interpretation. It was felt that minor discontinuities in unimportant sections were given as much consideration as the more important disclosures in critically stressed sections. The situation involves more than fear of the possibility of having this come true, for in some cases it actually happened. These situations resulted largely from the fact that the operating conditions and requirements applicable to the various casting members were not known to the inspectors or to the purchasers.

Such conditions have led casting producers to use caution in accepting unqualified orders calling for castings subject to radiographic examination. Orders of this type mean that the purchaser may radiograph any and all parts of the casting and perhaps reject the casting for some minor defect present in an unimportant section, such as the tie-in sections which carry no important stresses and which may be so designed that their factor of safety is extremely high.

To prevent occurrences of this type, the foundry management should request that the consumer submit marked drawings which designate the places that will be subject to radiographic examination. The thought underlying such a request is that marked drawings will help the foundryman materially in planning the production of the job. It will enable the operating department to provide for desirable metal feeding and in many cases reduce the use of inserts; in fact, the mold may be produced in a manner differing entirely from production procedures wherein no information as to casting requirements of important or highly stressed areas is given.

The foundry management likewise desires to discuss with the purchaser at the time the order is received the subject of interpretation of the radiographic results, since he desires to know if the purchaser has a set of standards that are reasonable and mutually satisfactory. If the purchaser has no standards, the casting producer will probably suggest that rejection be determined by a joint council of purchaser and foundryman.

There are some who believe that the product of the foundry industry is seldom, if ever, free from defects. In many cases, they have based this opinion on the continued appearance in the technical press of radiographs of defective castings. The fact that the authors usually state that the illustrations presented should not be used as an indication of the quality of castings in general apparently makes no impression.

It so happens that the radiographic methods are well adapted to the nondestructive examination of castings. Welded construction is likewise adaptable to this type of investigation, and it is believed that the technical literature contains as many reproductions of radiographs of defective welds as it does of castings.

Other metal forms have their share of defects also. It so happens that other methods of non-destructive testing are more adaptable, such as in the location of laminations and seams in plate and fabricated shapes.

RADIOGRAPHY AN AID TO DESIGNERS.

Foundrymen have great hopes that radiography will be of material aid in pointing out to designing engineers and purchasers in general that poor design is responsible for a large percentage of the castings that prove defective. For many years foundry engineers have requested that designers make their proposed designs more castable. They have continually suggested that the design engineer consult with them while the design is in the formative stage. If cooperation between designer, purchaser and foundryman existed at that time, many difficulties and much dissatisfaction could be avoided. It is believed that radiography will tend to bring about such cooperation because producers are being confronted with radiographs of castings which are not so designed that it is possible for the foundry to secure the quality of castings that the designer intends to have.

The foundryman today is also insisting to a greater extent that poor design features of castings be discussed with the purchaser. He realizes that the purchaser will reject a defective casting as shown by radiography, even if the designer has designed it so that it could be nothing else but defective. And even though the foundryman can convince the purchaser that poor design is responsible for the defect, the foundryman nevertheless has a defective casting on his hands.

Thus, there is a strong expectation that with the progress of radiography in the casting industry will come the appreciation by designers of more tolerant designing, since radiography makes it so easy for them to see the results of their handiwork.

If a casting is designed properly, the percentage of defects to be found resulting from manufacturing operations is low. Proof of this statement may be obtained at any foundry where a casting which has been standardized as to design and production details is being made regularly. Radiograph after radiograph will be found to be flawless.

RADIOGRAPHY AN AID IN MANUFACTURING.

Radiography is of value to many foundrymen as an aid in establishing the manufacturing procedure of a casting. A pilot casting may be completely radiographed in order to check on casting design and production methods. In this manner, changes can be made to the pattern or mold construction so that the next casting will be a perfect job. This prevents the repetition of a defect in a number of castings, all of which may be rejected.

It is believed that radiography is bringing about greater understanding of the service requirements of a casting. More attention is being given to the highly stressed areas of castings. The result of this has been that defects which previously may have caused rejection are now being repaired and accepted, since the sections in which they occur are not vital ones.

Also, purchasers are more willing to repair castings when they can reradiograph the casting and observe the condition of the repair. The Bureau of Engineering of the U. S. Navy has found that, because of radiography, it could authorize the repair of certain castings which would have been rejected before the advent of radiography.

RADIOGRAPHY IN ALUMINUM AND STEEL CASTINGS.

In the casting industry radiography has received more attention in the aluminum and steel casting branches.

Aluminum alloy castings such as those being used in the aircraft industry are being inspected by radiography in large numbers. One inspection service reports several thousand castings being inspected weekly by X-rays.

In the steel casting industry high-pressure high-temperature equipment is receiving the bulk of radiographic inspection, though considerable radiographic inspection is being given to structural castings which must operate in very exacting services. Gamma-ray radiography is being favored for the above types of inspection. The U. S. Navy has approximately 4800 milligrams of radium, split up into 19 capsules, for the inspection of castings of the above types purchased by the Navy. The Radium Chemical Co. of New York City has 34 capsules totaling 3800 milligrams of radium available for commercial radiography. Several of the more progressive foundries and users of castings have purchased radium and X-ray equipment for nondestructive testing.

High alloy content steel castings, such as the heat and corrosion resistant cast steels, are frequently inspected by radiography, the X-ray method being favored due to the relatively light sections.

It is believed that in view of the points disclosed above, the radiographic development in the casting industries is equivalent to that found in other industries. It may also be considered a factor in the technical development of the casting industries.

THE SECOND GROUP OF NATIONAL DEFENSE FEATURE TANKERS.

This article describes briefly the second class of National Defense Feature tankers. The tankers of the first class are slightly larger and have twin screws. Three vessels of this class are now in the Naval service as U. S. S. *Cimarron*, U. S. S. *Neosho*, and U. S. S. *Platte*. Reprinted from the April, 1940, issue Pacific Marine Review, published at 500 Sansome St., San Francisco.

The U. S. Maritime Commission expects shortly to issue contracts for the construction of six national defense feature oil tankers similar to six tankers which the Socony-Vacuum Oil Company are now building.

These Socony-Vacuum vessels are part of a U. S. Navy-U. S. Maritime Commission program for the construction of 24 high-speed tankers to be operated in regular service by American private operators and callable by the Navy in an emergency.

The first group of this program—12 twin-screw tankers built for the Standard Oil Company of New Jersey—is nearing completion. These vessels measure 11,500 gross tonnage and have 13,500 shaft horsepower.

The second group will consist of single-screw tankers with a gross tonnage of about 10,300 tons and 12,000 shaft horsepower.

The design for this second group of tankers was based on that of the tankers Mobilfuel and Mobilube, built for the Socony-Vacuum Oil Company by the Sparrows Point Yard of the Shipbuilding Division of the Bethlehem Steel Company.

The design was prepared under the direction of N. J. Pluymert, Naval architect, for Socony-Vacuum Oil Company, using the Bethlehem-Frear fluted bulkhead and longitudinal system of hull construction. The principal

characteristics of this size and type of national defense feature tanker are given in the table herewith.

This design is described in a paper on "Modern Tanker Design," read before the 1939 annual meeting of the Society of Naval Architects and Marine Engineers. We quote from this paper:

"The design has reduced the riveting to shell and main deck strake laps only, with the deck longitudinal riveted to avoid overhead welding, and the entire internal structure is welded.

PRINCIPAL CHARACTERISTICS.

Length over all, feet and inches	500–7½
Length B. P., feet and inches	487–6
Beam molded, feet and inches	68–0
Depth molded, upper deck, feet and inches	37–0
Draft molded, designed, feet and inches	29–8½
Draft, summer freeboard, feet and inches	30–0
Gross measurement, tons	10,300
Total displacement, tons	21,450
Total deadweight, tons	15,900
Tank capacity, 42-gallon barrels, barrels	129,000
Shaft horsepower	12,000
Speed on trial, knots	16½

Data given in this paper show that a tanker of the same size built several years back employed 1,058,976 rivets and 12,000 feet of welding. The Mobilube used 122,055 rivets and 371,168 feet of welding. (The new tankers will use about 80,000 rivets and 380,000 feet of welding, as the bottom plating is to be entirely welded.)

"A study of corrosion in the various members of the tank structure indicated the advisability of eliminating pockets and horizontal members where corrosion is accelerated by the collection of scale. The Bethlehem-Frear system was developed with the thought in mind of retarding corrosion as well as saving weight by the use of fluted bulkheads.

"The latest design with fluted bulkheads and a major use of welding shows a saving of nearly 1000 tons in the light weight of the vessel and its consequent increase in deadweight and earning power with a hull structure less liable to rapid deterioration or corrosion. This saving was made by a reduction of 800 tons or 18 per cent in hull steel and 200 tons in machinery. This saving in hull steel was not accomplished by using the minimum of permissible scantlings. The members at the top of the cargo tanks and the horizontal members carrying the maximum of 8/100 inch over classification scantling and the members in the center of the tank are as much as 4/100 inch over the minimum required scantling to allow for corrosion loss.

"In considering new designs of tankers, attention must be given to the maximum use of welding in conjunction with a design of internal structure of the cargo spaces which will eliminate as far as possible horizontal surfaces and pockets, thus reducing corrosion. The type of machinery must be carefully considered for the intended service so that full advantage is taken of the lightest combined weights of machinery and bunkers.

"World tanker construction shows a decided increase in both speed and cargo deadweight in tankers built in recent years. The use of lighter and more efficient machinery and the higher deadweight efficiency of recent welded designs provide for these increases with little change in the principal dimensions of the tanker."

Data with the paper indicate that on regular service Mobilube carries 15,687 tons of cargo at 14 knots on a fuel consumption of 27 tons a day.

On the trials of the Mobilube, the representatives of the Maritime Commission were very much impressed with the design of the tanker and the results obtained. After a conference, it was agreed that the technical staff of Socony-Vacuum would adapt the design to assure 16½ knots sea service speed.

It was figured that this speed would require 12,000 shaft horsepower after the lines at the bow had been fined somewhat to give less resistance without unduly sacrificing cargo capacity.

The Mobilube has 4400 shaft maximum horsepower, and on her trials made 14 knots speed. The designers were able to get 12,000 shaft horsepower on the new tankers in the same dimensions of engine room used on the Mobilube.

Two boilers of sufficient capacity with cross-compound, double-reduction geared turbine developing three times the horsepower of the slower speed design.

The machinery and systems accessory to the power plant requiring increase in capacity and power included: the condenser, circulating pumps and condensate pumps, air ejectors and inter and after condensers, feed pumps and deaerating feed heater, fuel oil and lubricating oil systems, and evaporator plant.

Auxiliary machinery layout and the entire electrical system are to be identical with the Mobilube.

This procedure in design has developed a combination of commercial and national defense types that will be satisfactory as a naval fleet tanker with a minimum sacrifice of commercial features.

The cargo oil pumps installed will discharge the entire cargo of 133,000 barrels of oil in approximately 14 hours.

HIGH PRESSURES AND TEMPERATURES PLANNED FOR MARINE FIELD.

The construction program of the U. S. Maritime Commission includes four high-pressure installations, one employing 1200 pounds steam pressure and 950 degrees F. total steam temperature operating on the straight Rankine cycle and three using 1200 pounds and 750 degrees F. steam with the regenerative reheat cycle. This interesting article by J. E. Schmeltzer, Asst. Director, Technical Division, U. S. Maritime Commission, is reprinted from the April, 1940, issue of Combustion, published by Combustion Publishing Co., Inc., 200 Madison Ave., New York.

One often reads of the American clipper ships which dominated our merchant marine a century ago. That these sailing ships were so successful in capturing and holding the shipping business of those days against foreign competition appears to have been due largely to their superior speed and their operating economy.

Taking a leaf from the record of these past experiences, the U. S. Maritime Commission has embarked upon a program of design and construction of a large number of cargo vessels to operate at speeds several knots faster than has been our previous practice with such vessels and with operating economies, as concern fuel expenditure per ton-mile, fully one-third better than the ships constructed twenty years ago. The present program involves about 200 ships

of which 21 have been completed and 120 are under construction. Of these, over two-thirds are propelled by steam and the remainder by Diesel engines. They include cargo ships, combination cargo and passenger ships and a few passenger liners.

STEAM CONDITIONS ADVANCED FOR ECONOMY.

To accomplish the desired operating economy it has been necessary to employ higher steam pressures and temperatures and many refinements in power plant layout, adapted to marine conditions, such as have proved most effective in land practice. While 450 pounds steam pressure and 750 F. total steam temperature have been adopted for most of the cargo ships, the Maritime Commission has given evidence of its progressive spirit by planning to employ steam conditions of 1200 pounds, 950 F. on one cargo ship and 1200 pounds, 750 F., with the regenerative reheat cycle, on another cargo ship, for the American Export Line, and on two large 25-knot trans-Pacific liners of 80,000 shaft horsepower for which specifications have been issued and on which bids are now being asked. It is the opinion of the Commission's Technical Division that the experience to be gained with these high-pressure high-temperature installations will eliminate the necessity of advancing by steps through steam conditions in the intermediate range—a procedure that is usually costly.

The C-2 type of cargo ships, of approximately 9700 tons deadweight and 6000 shaft horsepower, designed for 15½ knots, employs 450 pounds steam pressure and 750 F. total steam temperature with cross-compound turbines and electrically driven auxiliaries. Each has two oil-fired water-tube boilers having a guaranteed efficiency of at least 87 per cent, and operated in all cases with automatic combustion control for which several types are being tried out. Two or three stages of feedwater heating are employed, according to whether economizers or air preheaters are provided. The feed system is closed and deaerating heaters are used for removing oxygen from the boiler feedwater. As in the practice with many of the later stationary installations, the boilers and the turbine are located in the same compartment in order to shorten the piping and to permit better observation by those in charge of operation.

Although designed for an overall fuel consumption of 0.60 pound of oil per shaft horsepower, the first ship of this class to go into commission did 0.545 pound on trial which is equal to 0.575 pound per shaft horsepower-hour for all purposes.

The C-3 type is somewhat larger and of 8500 shaft horsepower, with a speed of 16½ knots. It is one of these ships for which a study has been made for an installation to operate under steam conditions of 1200 pounds, 950 F. at the superheater outlet, operating on the Rankine cycle. A fuel consumption of better than 0.5 pound per shaft horsepower for all purposes is anticipated under such conditions.

The reheat cycle, as adopted with 1200 pounds, 750 F. for the one cargo ship and two liners mentioned, offers a decrease of 12 per cent in fuel consumption over 450 pounds, 750 F. in the case of the 8000-HP. installation and approximately 15 per cent for the 80,000-HP. plant. The smaller of these installations will consist of two high-pressure boilers each having one superheater of the convection type and a separately fired reheater, located within the boiler setting. The only high-pressure steam line will be that from the superheater to the turbine throttle, which is a relatively short distance, all other piping being for 450 pounds or less. Steam for all the auxiliaries will be supplied at 450 pounds. The turbine will be of the triple-cylinder type having the high and intermediate sections in tandem.

REHEAT AND RANKINE CYCLES SHOW SAME PERFORMANCE UNDER CONDITIONS SELECTED.

The calculated performance for 1200 pounds, 750 F. with reheat is approximately the same as for 1200 pounds, 950 F., without reheat, and it has the advantage of not requiring materials to withstand the higher temperature. Because of this it is not expected that such a plant will cost much more than that of a 450-pound, 750 F. installation. Furthermore, the weight is not likely to be any greater for the cargo ship and will actually be much less per shaft horsepower for the two 80,000-HP. liners. This is because of smaller high-pressure piping, a reduction in the size of turbine and a reduction in the size of certain auxiliaries due to the better steam consumption and to less fuel burned.

An additional advantage of the reheat cycle is ability to start the main units more quickly than if the higher steam temperature were employed, as the reheat can be cut out during starting. Its only purpose is to improve economy during steady operation.

The large trans-Pacific liners will each have three evaporating plants with a combined capacity of 90,000 gallons per day. These will use steam bled from the main turbines at 5 pounds gage. In order to keep down the amount of makeup the soot blowers will employ air instead of steam. These will be operated continuously so as to avoid soot deposits on the decks such as would be likely with intermittent blowing. Air conditioning, refrigeration and lighting on these ships account for a very substantial auxiliary load.

Comparing the steam-driven ships with those propelled by Diesel engines it may be pointed out that the former employ a somewhat lower grade of oil than the latter; so that, from the standpoint of fuel cost, the economies of the two types are very close. However, recent developments in Diesel engines, embodying reduced weight and space per horsepower through employment of higher speeds, have accentuated the problem of having a suitable fuel. Since the operator of a Diesel cargo ship often has to purchase its fuel oil in foreign ports where he has no control of the quality, it is most desirable to employ engines that will economically burn oil of a wide range in quality if high maintenance and excessive operating costs are to be avoided.

Furthermore, in this country, the employment of improved refining processes for the extraction of gasoline is reducing the available amount of distillate such as is suitable for Diesel engines and the domestic demand for distillate fuel is increasing with an accompanying increase in the market price—a factor that must not be neglected in considering the probable fuel costs for a Diesel installation over its useful life, which may be considered twenty years. On the other hand, the fuel remaining after removal of gasoline and other distillates from the crude is usually suitable for burning under steam boilers.

Therefore, it is most important that motorships be propelled by engines capable of burning low grade fuel oil of a quality and price not too much in excess of that which can be burned in steam plants; otherwise the motorship, in general, will find it difficult to compete with steam.

In conclusion, it may be pointed out that while marked advances are being made in the marine field, through adoption of many practices and refinements that have been worked out satisfactorily on land, the marine engineer must ever keep in mind simplicity and reliability and the particular conditions imposed through operation at sea.

SUPERPOSED TURBINE BLADE RESEARCH.

Abstract of a paper presented at a meeting of the Philadelphia Section, Philadelphia, Pa., February 27, 1940, of the American Society of Mechanical Engineers, by F. T. Hague, Manager, Engineering-Steam Division, Westinghouse Electric & Manufacturing Co., South Philadelphia, Pa. Reprinted from April, 1940, issue of Mechanical Engineering, published by the American Society of Mechanical Engineers, Easton, Pa.

Superposed turbines pass from 1 to 1½ million pounds of steam per hour at a density five times as great as that in the low-pressure turbines with which they are in series, and the steam temperature is 900 F. The passage of a large volume of dense high-temperature steam through the first wheel creates conditions unlike those with which there has been much previous experience. For each 20 F. increase in temperature in this region, the plastic flow or creep rate of steel is doubled and other physical characteristics are adversely affected.

To obtain good economy at partial loads the incoming steam is fed to this first wheel at only a portion of its periphery, and the blades passing into and out of this steam jet receive a most violent shock each revolution of the turbine. To form a mental picture of this shock imagine an impulse blade moving 350 MPH. abruptly entering a dense steam jet moving 1200 MPH. Shock and vibration of these first-row blades are aggravated by this intermittent steam loading. This intermittent steam loading has been conventional for many years on all sizes of condensing turbines. It has been only on those few machines of the large superposed class that its use, in an endeavor to develop the ultimate in partial load efficiency, has caused some blades to break at their root where they are fastened to the spindle. The prompt failure of three successive sets of physically larger blades on a large super-

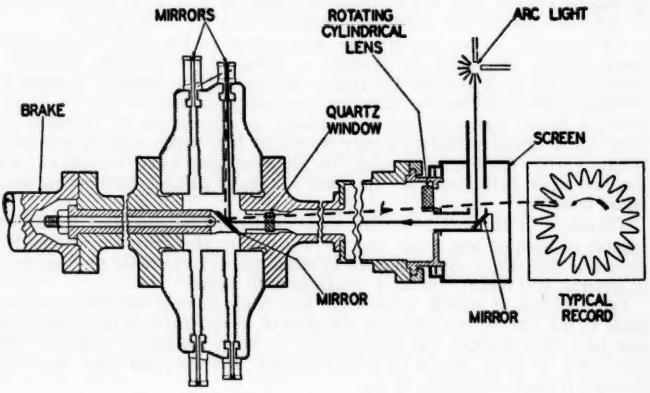


FIGURE 1.—OPTICAL SYSTEM, IMPULSE-BLADE RESEARCH.

posed turbine indicated that something more than brute strength is required of impulse blades in this application when intermittent steam loading is involved.

A research was initiated to develop a better understanding of the conditions affecting the operation of impulse blades in superposed turbines. Nothing short of actually photographing the blades' movements under actual operating conditions promised to give adequate and accurate records of the stresses imposed on these blades. An ingenious optical method, Figure 1, of accomplishing this has been devised and the turbine blades are made to write their own story in a camera adjacent to the turbine. Tiny mirrors are mounted inside eight different blades in the first and second blade rows. A light beam projected through a hole in the turbine shaft is reflected by an inclined mirror to the mirror in the whirling blade being photographed and then back out through the shaft to the stationary camera or viewing screen. By taking pictures every one-half second on moving film, an exact record of blade performance is obtained for any desired set of operating conditions. The turbine optical system is so proportioned that the movement of the impulse blade is magnified 250 times by the movement of the light spot.

The first development turbine with this built-in optical system has been in operation in our plant since September, 1939. More than 20,000 photographic records have been obtained showing the performance of impulse blades in a wide range of operating conditions. Single blades and shrouded blade groups have been tested in the first and second rows of a Curtis stage under available plant steam conditions of 400 pounds per square inch and 600 F. Although the pressure and temperature of our plant are not those met in commercial service, nevertheless the photographic method developed will apply with equal success to any commercial steam conditions.

THE EVIDENCE OBTAINED.

Examination of a typical photographic record of a blade's movements during one complete revolution of spindle shows that the blades vibrate like a tuning fork, Figure 2. The vibration is increased as a blade enters the

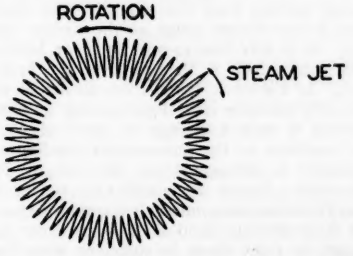


FIGURE 2.—TYPICAL BLADE VIBRATION IN RESONANCE.

steam jet. The blade vibrates while in the jet and continues to vibrate with slowly decreasing amplitude after it leaves the steam jet. This slowly decreasing amplitude of vibration is significant because it is caused by the internal molecular friction of the blade material, i.e., the damping characteristic of the blade steel. The blades vibrate at their natural frequency. This is true whether the steam shock is light or strong and is independent

of turbine speed. The natural frequency of vibration is fixed by the blade design and material.

In superposed turbines the frequency of vibration of first-row blades is about 2500 cycles per second, one of the highest notes on a piano, or 200,000,000 cycles per 24-hour day. Blades stressed at their fatigue limit usually break within 10,000,000 cycles. This explains why failures of blades in the first row of superposed turbines usually develop quite promptly, if they fail at all, particularly if the turbine is loaded at half load or below where intermittent steam loading is involved. As the turbine load is increased the steam jet covers more and more of the periphery of the first wheel until, above three-quarters load, steam is applied throughout the entire periphery of the first wheel. When the full-admission condition is established, this vibratory stress condition is eliminated.

WHAT THE STUDIES SHOW.

As in diagnosing a new ailment, it is important to know which factors affect the condition and which do not. The optical test studies and practical experience have shown both.

Strange though it may seem, the amount of load on the turbine has relatively little to do with the maximum blade stress. The factors which influence this condition are mechanical resonance and the damping characteristic of the blade steel.

Mechanical resonance involves a relationship of two frequencies, and in turbines one of these is the frequency of the shock from the steam jet. The other is the natural vibrating period of the blade. When these two frequencies are in such a relation that one is an exact integral multiple of the other, resonance results. When resonance is produced in any mechanical system, the magnification of stress is limited solely by the damping of the material used. The ability of a turbine blade to operate without breakage depends largely on whether the damping within the blade is sufficient to choke the vibration established during resonant conditions.

RESONANCE.

With any constant turbine load, the magnitude of blade stress has been found to vary over a four-to-one range as the turbine speed is changed as little as 2 per cent. It is this four-to-one range or build-up of blade stress at constant turbine load which is the cause of the breakage of blades in superposed turbines. In Figure 3, taken from an actual turbine in constant-load operation, the only variable is a slight change in turbine speed; in this particular test turbine it took a change in speed of 1⅓ per cent to pass from the resonant condition to the nonresonant condition.

"Out of resonance" is illustrated on the left of Figure 3 and "in resonance" on the right. Figure 3 consists of seven or eight photographic records of blade performance taken one-half second apart as the turbine speed is slowly changed from 3090 to 3130 RPM. In order to get all of these records on one chart to show them in sufficient scale they are overlapped and for sake of clarity all of the overlapping records on the first and last have been eliminated. The first and last records represent "out of resonance" and "in resonance" blade performance.

In the "out of resonance" record, Figure 3, when the blade passes the steam jet, the steam jet hits the blade out of phase with its existing motion and a new order of frequency is established. It is that continual hitting of the blade out of phase with its existing motion as it passes through the steam jet that prevents the blade from building up its vibration to a large amplitude and its stress to a large value.

In contrast to this, the record on the extreme right of Figure 3 indicates that as the blade passes through the steam jet there is no variation whatsoever in the frequency. In that case conditions required for resonance have been established. The steam jet is hitting the blade in phase with its existing vibration and, as a result, vibration amplitude is built up to a large value, controlled and limited by the damping capacity of the blade steel. It is that condition which increases the blade stress by a factor of four to one, even though the turbine load is constant, and it is that amplification of blade stress which breaks the blades in superposed turbines.

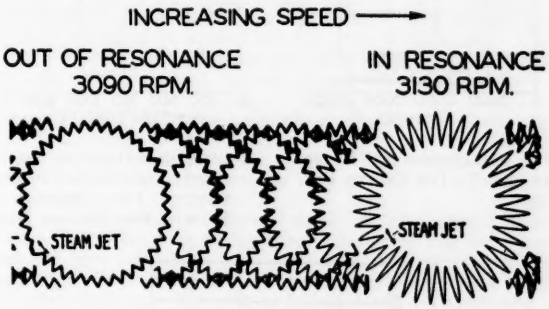


FIGURE 3.—TYPICAL RECORDS OF BLADE MOTION AT CONSTANT LOAD.

It is obvious, therefore, that the impulse blading of large superposed turbines must be designed to operate at safe stresses under resonant conditions. Although actual turbines run at constant speed, the natural period of blades will vary more than 10 per cent by virtue of manufacturing variations, changes in operating temperatures, and the like, so that some blades will be in resonance under all operating conditions.

DAMPING.

One phase of the investigation has been to explore the factors affecting damping. Several important damping relationships have been discovered.

1. Damping is not constant but increases with blade stress and is extremely low at low stress, Figure 4. It is as important to know this relationship as to know the strength of steel.

2. The damping characteristics of different steels vary widely. The widely used 12 per cent chrome steel has more damping in its usable application range than any other material yet investigated. The materials of the chrome-nickel class that excel in high strength at high temperatures have about one-tenth as much damping as 12 per cent chrome steel.

3. Damping becomes less as the frequency of vibration increases, Figure 5. This means that the increased stiffness of heavier blades does not result in much lower blade stresses.

4. Damping decreases rapidly at higher operating temperatures, particularly beyond 750 F. At 100 F. the damping is only one-third as much as at room temperature, and is decreasing precipitously, Figure 6.

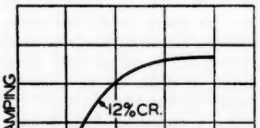


FIGURE 4.—DAMPING OF BLADE STEELS, 700 F., 750 CYCLES PER SECOND.

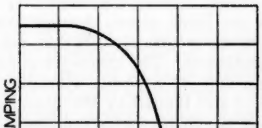


FIGURE 6.—DAMPING OF 12 PER CENT CHROMIUM STEEL, 750 CYCLES PER SECOND, 20,000 POUNDS PER SQUARE INCH.

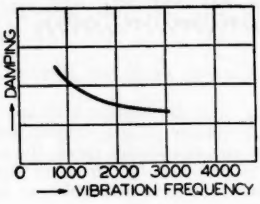


FIGURE 5.—DAMPING OF 12 PER CENT CHROMIUM STEEL AT 1000 F.

SEPARATE DAMPERS.

These findings suggest the possibilities of applying some sort of separate dampers to the blades. We are investigating separate damping means that increase the damping of blades by five to ten times. A steel capsule, partly filled with material that is fluid at operating temperatures, can be inserted inside a blade and maintain its full damping capacity up to 1500 F.

We do not mean to infer that superposed turbines for present-day standards of temperature and pressure must have separate blade dampers to operate safely. This research merely points out that such means would greatly increase the factor of safety of impulse blades and such means may be required to meet future operating conditions.

Our research shows that impulse-blade stresses can be calculated. The calculation system developed by R. P. Kroon based on this resonant method of analysis permits a rational evaluation of controllable design features and makes it possible to predict whether a certain design is safe.

EXPERIMENTAL TURBINE AT SCHUYLKILL.

Building on the experience with this experimental turbine at South Philadelphia, a second one with full-size impulse wheel, duplicate of the

impulse stage of the 50,000 Kw. West Penn unit, was built for research study of blade performance under 1250 pounds per square inch, 900 F. steam operating conditions. This turbine with its 10,000 Kw. generator and 15,000 HP. water brake began testing operation in February, 1940, in the Schuylkill plant of the Philadelphia Electric Company. This turbine takes steam from the station 1250-pound steam lines and exhausts into the low-pressure 240-pound lines, permitting its operation as a superposed unit without interference with the station's regular operation. This turbine is built solely for research purposes and has many special constructional features, including single and double blades, separate blade dampers, different shroud attachments, and various blade groupings. The optical system will record the performance of sixteen different blade combinations. Two rotors are being built with different blade designs and the presence of the 10,000-Kw. generator will permit life tests by loading into the Philadelphia Electric system.

Reference to this unit must acknowledge the joint research relationship with the Philadelphia Electric Company, who are making available the use of the station, operators, and steam supply. This contribution to research on a central-station industry problem is a tribute to the broad viewpoint of the management.

Today we are making a progress report on our research studies. To maintain proper perspective it should be remembered that the impulse-blade problem has been critical on only a few large superposed turbines, which, although an important factor in the industry, are nevertheless a small portion of the turbine business. We have an excellent tool for diagnosing the duty and stress of impulse blades. From the research studies of this installation will come a visual record of impulse-blade operation which will expedite the engineering solution of the impulse-blade problem. It has been our experience that research problems are more readily solved when measuring instruments are evolved which make possible quantitative measurements of occurring phenomena. We have that instrument in this photographic method.

LOAD TESTS OF A COMBUSTION GAS-TURBINE.

The reader is referred to the Notes of this JOURNAL for August, 1939, for an article on the history, development and prospects of the combustion gas turbine by Dr. Adolph Meyer. The 4000-Kw. set for the standby power station of the town of Neuchâtel, Switzerland, the construction of which was touched on in that article, has now been very thoroughly tested under the guidance of Dr. A. Stodola. The results attained have realized every expectation and opened the way to further developments of the combustion gas turbine. The following account of this test is abstracted from an article by Dr. A. Stodola appearing in the April, 1940, issue of the Brown, Boveri Review, the house journal of Brown, Boveri & Company, Ltd., of Baden, Switzerland.

The unit tested is a gas-turbine set built by Brown, Boveri & Co., Ltd., of Baden, comprising a combustion turbine and consisting only of the essential components, such as compressor, combustion chamber, turbine (exhausting directly to atmosphere) and generator. The deliberate omission of any additional refinement, such for example as heat exchanger between the exhaust gases and the combustion air, is justified by the fact that the

turbine is intended for a bombproof plant which will be in service only a few hours per annum, so that an increase in cost for the purpose of raising the efficiency would not be economical.

In spite of the simplicity of the arrangement, the tests showed the efficiency referred to the heat contained in the fuel, and the heat equivalent of the electrical output of the generator to be 17.38 per cent, representing an efficiency which, together with the many constructional advantages of the set, renders it in many cases competitive. A feature specially worthy of note is the fact that the plant requires no cooling water.

The present-day possibilities of realizing a successful gas turbine, resulting from the improvement of the compressor efficiency on the one hand, and the availability of heat-resisting materials on the other, would have appeared unthinkable but a short time ago. The axial type compressor developed some seven years ago by Brown Boveri in connection with the Velox steam generator, enables an improvement in the performance of the gas turbine cycle of 65 per cent, compared with the best result attainable with a centrifugal type of compressor, while a further improvement of 18 per cent is rendered possible by the increase in temperature which can be withstood for a given creep limit by the best heat-resisting steels now produced by the metallurgists.

Were it not for these developments, the only possibility for the realization of the gas turbine would be that appreciated by Holzwarth already in 1905, namely the explosion cycle, and which was consequently persistently pursued by him in spite of many sacrifices, until in 1933, in collaboration with Brown, Boveri & Co., a working explosion turbine plant was produced. The en-

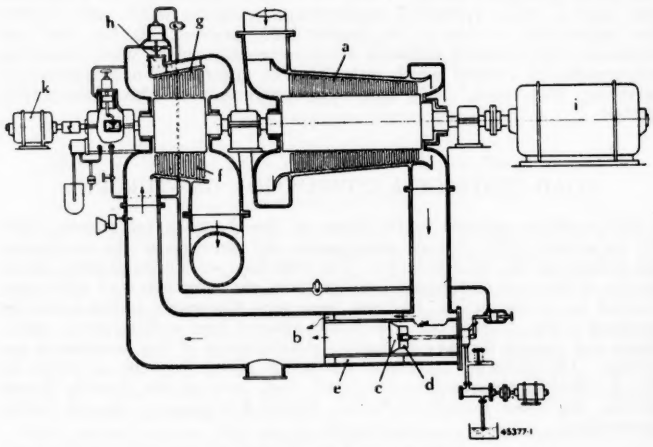


Fig. 1. — Diagram of the simplest form of a combustion turbine plant for oil fuel with a reaction type gas turbine and an axial compressor.

a. Axial compressor.
b. Combustion chamber.
c. Burner nozzle.
d. Air nozzle.
e. Cooling air ring.
f. Gas turbine blades.
g. Gas turbine.
h. Relief valve.
i. Generator.
k. Starting motor

couraging results obtained with this unit led about a year and a half ago to the placing of an order by the Aug. Thyssen steel works in Hamborn for a 5000-Kw. set.

Brown, Boveri & Co. have during the course of development of the Velox boiler and the supercharging sets devoted considerable attention to the scientific investigation of fluid flow conditions, and to the solution of constructional problems of gas turbines and axial blowers. The result of this development work is seen in the plant described below, the individual efficiencies of the components representing the highest values hitherto achieved.

In the diagrammatic arrangement Figure 1, *a* is the compressor, which compresses the air to about 3—4 kg/cm^2 abs. The drum is secured by shrinking and welding on to the shaft ends. The shape of the inlet passage, as has been shown by model test, enables the kinetic energy of the gas stream to be converted into pressure. Guide blades give the air entering the combustion chamber *b* a slight rotational motion. Only part of the total air quantity is made use of for combustion, so that the temperature reaches about 2500 degrees F. The remainder of the air subsequently mixes with the gases after combustion is completed.

The turbine is provided with a multi-stage reaction blading. The rotor, Figure 2, consists of two shaft ends, each formed in one piece with the end disc, and of an intermediate unbored disc of constant strength section, welded at its periphery to the two outer discs, in accordance with the well-known process of the manufacturer. This process has, apart from the advantage of the unbored disc of constant strength, the additional advantage of doing away with the necessity of shrinking the discs, or of providing keys and keyways required in the case of a continuous shaft, and which are undesirable where high temperatures are encountered. The rotor is built up of easily forged components and the large distance of the welds from the axis results in very low stresses in the weld material, due to bending or possibly vibration of the rotor. The process has proved to be entirely reliable and given ex-

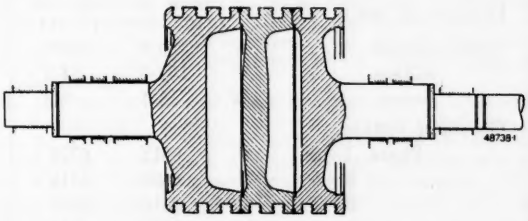


FIGURE 2.—CROSS SECTION OF THE BROWN BOVERI TURBINE ROTOR.

cellent results in numerous steam turbines built by this firm. The difference between the axial thrusts of the turbine and the compressor is taken up by means of a common segmental thrust bearing.

The thermal expansion on an overall length of the turbine of 2 m, measured over the bearings, and with an initial gas temperature of 1000 degrees F., is appreciable but the difference between the expansions of the casing and the shaft, however, due to the insulating effect of the lagging, is quite small. The glands consist of thin alloy steel strips caulked into grooves in the shaft.

Table 1: TEST RESULTS.

Item	Test No.		I	II	III
	Load		Light	abt. 4000 kW	abt. 3000 kW
1	Date of test		7. 7. 39	7. 7. 39	7. 7. 39
2	Duration of test . .	min	15	60	30
	Fuel measurements.				
3	Kind of fuel oil . .			gas oil	
4	Moisture content . .	%		0·0	
5	Ash content	%		0·0	
6	Elementary analysis C	%		86·80	
	„ H	%		12·45	
	„ S	%		0·66	
7	Net calorific value	kcal/kg		10143	
8	Gross calorific value	kcal/kg		10846	
9	Density at 68° F . .			0·851	
10	Density at 77° F . .			0·848	
11	Density at 86° F . .			0·845	
12	*Corrected fuel consumption of the gas turbine . . . . .*	lb/h	*1812*	*4338*	*3649*

Table 2: ELECTRICAL READINGS.

Test No.		II	III
Duration of test . . .		from to 10^{10} 11^{10}	from to 11^{25} 11^{55}
Exciter-current	A	130·8	108·4
voltage	V	102·7	84·9
power	kW	13·4	9·2
Generator voltage:			
Phase T—S	V	6611	6724
R—S	V	6580	6716
R—T	V	6616	6691
average	V	6602	6710
Generator current:			
Phase R	A	363	270
S	A	343	256
T	A	349	261
average	A	352	262
Power factor		0·998	0·999
Output from	kW	4015	3041
Wattmeter readings . .	kWh	4015	1521
Output from	kW	*4021*	*3057*
Watthourmeter readings	kWh	*4021*	*1529*

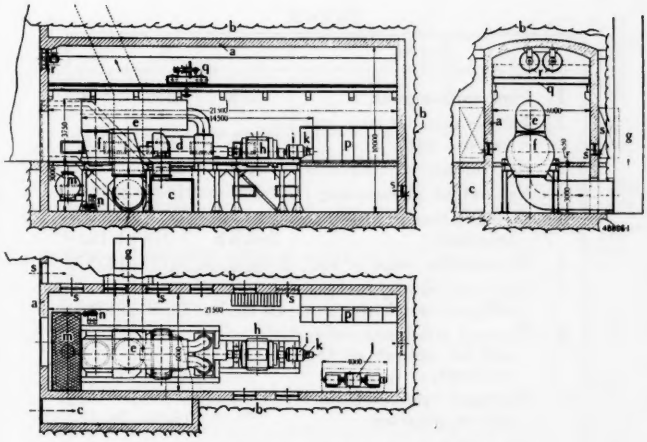


Fig. 3. — Underground gas turbine plant for an output of 4000 kW.

a. Enclosing building.
b. Excavation in the rock.
c. Combustion air inlet duct.
d. Air compressor.
e. Combustion chamber.
f. Gas turbine.
g. Exhaust gas duct.
h. Electric generator.
i. Starting motor.
k. Exciter.
l. Starting Diesel unit.
m. Fuel tank.
n. Fuel pump.
p. Control board.
q. Crane.
r. Fan for room ventilation.
s. Ventilating air inlet.

The load and governing tests were performed on July 7th, 1939, at the maker's works in Baden under my personal supervision, with the assistance of the officially recognized Swiss Association of Steam Boiler Proprietors and of the test laboratory staff of the Swiss Association of Electrical Engineers for the power measurements. The temperature and pressure measurements were taken by the firm's personnel. The calorific value of the fuel was determined by the fuels department of the Swiss Federal Material Testing Laboratories.

The fuel consumption was determined by weighing in a large vessel on a calibrated sliding-weight balance, the vessel containing sufficient fuel for more than an hour's run. Readings were taken every ten minutes. The electrical output at the terminals of the three-phase generator was measured by means of a precision Kwh.-meter read every two minutes. The results of the measurements and the characteristics of the fuel are given in Tables 1 and 2.

EFFICIENCY OF THE PLANT.

In order to obtain with the ambient temperatures of 77.5 and 78.8 degrees F. prevailing during tests II and III, the same compression and expansion ratios and efficiencies as would obtain at a temperature of 68 degrees F., on which the guarantees were based, the speed was raised during tests II and III from 3000 to 3020 and 3030 RPM. respectively. The readings then recorded are given in Table 3.

These figures show that the performance of the unit is better than was guaranteed. It is, however, interesting to analyze these results with the aid of the pressure and temperature readings. The more important values are set out in Table 4.

Table 3.

	Test No.		II	III
1	Fuel consumption . . .	lb/h	4338	3649
2	Temperature at the gas turbine inlet	°F	1025	916
3	Net output at terminals	kW	4021	3057
4	Net output at coupling .	kW	4184·3	3193·8
5	Consumption per kWh at terminals	lb/kWh	1·078	1·193
6	Net calorific value of fuel	B.Th.U./lb	18257	18257
7	Heat supplied by fuel per kWh at terminals . .	B. Th. U.	19680	21780
8	Thermal efficiency referred to output at the terminals	%	17·38	15·67
9	Efficiency referred to output at coupling . . .	%	18·04	16·37
	or, for the guaranteed conditions, namely: air temperature 68° F, speed 3000 r. p. m., the figures are: —			
10	Output at the terminals	kW	4000	3026
11	Output at the coupling	kW	4163	3162
12	Temperature at the gas-turbine inlet	°F	998	890
13	*Thermal efficiency referred to terminal output* . .	%	*17·38*	*15·67*
14	Thermal efficiency referred to coupling output	%	18·04	16·37
15	Fuel consumption, based on fuel having a net calorific value of 18000 B. Th. U./lb	lb/kWh	1·090	1·210
16	Guaranteed consumption based on the net calorific value of 18000 B. Th. U./lb	lb/kWh	1·164	1·260

In order to determine the average temperature of the products of combustion at the inlet of the gas turbine with greater accuracy than is possible by means of local measurements by thermo-couples, measurements of the exhaust gas quantity were made in the preliminary trials by means of a standard nozzle located in the exhaust duct, and which was subsequently removed in order to avoid any unfavorable effect on the efficiency by the additional pressure drop. The air quantity delivered by the compressor was then given by deduction from the total gas quantity of the weight of the fuel, thus enabling the speed-volume characteristic of the compressor to be

Table 4: PRESSURES, TEMPERATURES AND AIR AND GAS QUANTITIES.

Test No.			I	II	III
Air compressor:—					
Abs. pressure at compressor inlet	p_{1c}	lb/sq. in.	14·056	14·056	14·056
Abs. pressure at compressor outlet	p_{2c}	lb/sq. in.	53·72	61·69	60·26
Temperature at compressor inlet	t_{1c}	°F	73·8	77·6	78·8
Temperature at compressor outlet	t_{2c}	°F	357·8	397·2	393·4
Combustion chamber:—					
Fuel quantity (calorific value = 18,257 B. Th. U./lb)	G_F	lb/h	1,812	4,338	3,649
Fuel temperature	t_F	°F	75·2	75·2	77
Gas turbine:—					
Pressure drop in stack (calculated)	ΔP_{Stk}	in. WG	0·866	1·102	1·024
Abs. pressure at turbine inlet	p_{1t}	lb/sq. in.	52·80	60·77	59·31
Abs. pressure at turbine outlet (barometric pressure $+\Delta P_{Stk}$)	p_{2t}	lb/sq. in.	14·21	14·22	14·215
Temperature at gas-turbine inlet	t_{1t}	°F	705·2	1,067	987·8
Temperature at gas-turbine outlet	t_{2t}	°F	325·4	532·4	487·4
Generator:—					
Terminal output	L_{term}	kW	0	4,021	3,057
Generator losses	L_{GL}	kW	0	161·3	136·8
Coupling output	$L_{C\,Gen}$	kW	0	4,184·3	3,193·8
Air volume delivered by compressor:—					
Speed	n	r. p. m.	3,020	3,020	3,030
Air volume (from speed-volume curve)	V	ft³/h	6,992,330	6,992,330	7,027,650
Air density at inlet ($\gamma_1 = p_{1c}/RT_{1c}$)	γ_1	lb/ft³	0·0710	0·0706	0·0704
Weight of air $G_A = \gamma_1 V$	G_A	lb/h	497,808	493,838	494,400
Weight of gases $G_G = G_A + G_F$	G_G	lb/h	499,620	498,176	498,049
Temperature at gas turbine inlet, calculated from fuel quantity:—					
Heat supplied with air per mol.* at temperature t_{2c} (from physical tables)	H_1	B. Th. U./mol	5,688	5,940	5,920
Net calorific value of fuel	H_u	B. Th. U./lb	18,257	18,257	18,257
Heat supplied by fuel per mol. of furnace gases $H_2 = \frac{G_F \cdot H_u}{G_G} \cdot m$ ($m = 29·0$)	H_2	B. Th. U./lb	1,921	4,608	3,883
Total heat supplied $H_1 + H_2$	H_{tot}	B. Th. U./lb	7,609	10,548	9,803
Temperature at turbine inlet	t_{1T}	°F	627·5	1,025·5	917·5
Gas quantity from flow formula:—					
$G_G = C \frac{p_{1t}}{\sqrt{T_{1t}}} \sqrt{1 - \left(\frac{p_{2t}}{p_{1t}}\right)^2}$	G_G	lb/h	498,248	495,602	501,996
Air Quantity $G_G - G_F$	G_A	lb/h	496,436	491,264	498,347
Compressor power:—					
Compression ratio	$\frac{p_{1c}}{p_{2c}}$	—	3·82	4·38	4·28
Adiabatic drop	H_{ad}	ft.	46,260	52,493	51,673
Adiabatic power	L_{ad}	kW	8,650	9,710	9,660
Adiabatic temperature rise	Δt_{ad}	°F	245·7	277·2	273·6
Measured temperature rise	Δt_m	°F	286·4	320	315·6
Efficiency from temperatures $\eta_{ad_t} \cdot \frac{\Delta t_{ad}}{\Delta t_m}$	η_{ad_t}	—	86·4	86·6	86·9
Coupling efficiency $\eta_{ad_c} = \eta_{ad_t} - 2$ %	η_{ad_c}	%	84·4	84·6	84·9
Power at coupling	L_{cc}	kW	10,250	11,480	11,380
Gas turbine power:—					
Expansion ratio	$\frac{p_{1c}}{p_{2c}}$	—	3·715	4·27	4·17
Adiabatic drop	H_{ad}	ft.	63,976	95,208	87,206
Adiabatic power	L_{ad}	kW	12,000	17,725	16,460
Coupling power $L_{CT} = L_{CC} + L_{CGen}$	L_{CT}	kW	10,250	15,664·3	14,573·8
Efficiency from power ratio $\frac{L_{CT}}{L_{ad}}$	$\eta_{ad\,T}$	%	85·4	88·4	88·4
Total efficiency of compressor and turbine:—					
$\eta_{tot} = \eta_{ad\,C} \cdot \eta_{ad\,T}$	η_{tot}	%	72·1	74·8	75·0

* The mol. is here taken as a weight numerically equal to the equivalent molecular weight m in lb. of the gas mixture.

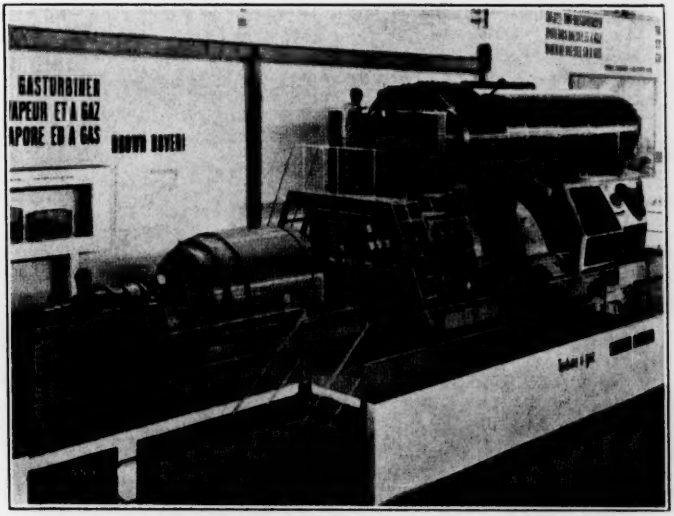


FIGURE 4—THE 4000-KW. GAS TURBINE SET ON SITE OF ERECTION.

drawn. For the gas turbine, the following approximate flow formula was employed:

$$G = C \cdot \frac{p_1}{\sqrt{T_1}} \sqrt{1 - \left(\frac{p_2}{p_1}\right)^2}$$

the constant factor being determined by means of a calibrated nozzle. The inlet temperature was then derived from the heat balance equation of the combustion chamber, radiation losses being neglected. Knowing the temperatures, the individual efficiencies can also be determined. The pressure of the compressor suction flange was slightly below atmospheric, as local conditions recessitated the provision of a wooden duct for the air supply. An allowance of 2 per cent was made to cover bearing friction, gland leakage and heat loss by radiation. Owing to a source of error in the measuring arrangements of the test, the loss of head in the exhaust duct had to be estimated analytically. This loss of head, with a length of duct of about 15 meters of 1800-millimeter diameter, 9 meters of 2200-millimeter diameter, 15 meters difference of level and three bends, provided with deflecting vanes, amounted—as was calculated after deduction of the natural draft—to 1.102 inches wire gauge. This value was calculated as accurately as possible, because the conditions on site will be approximately the same as those obtaining during the test, so that no correction of the guarantees is necessary in this respect.

GOVERNING TESTS.

During load changes the amount of fuel is regulated directly by the governor which allows of variations over a range of no-load to full load and vice versa. The speed variation during these trials was read directly on the tachometer scale, the results being as follows:

Unloading from full-load to no-load:
Momentary speed increase: 5.8 per cent after 22.5 seconds.
Permanent speed increase: 2.5 per cent after 1 minute 48 seconds.

Applying load from no-load to full-load (measured by Brown Boveri personnel):

Momentary speed drop: 5 per cent after 33 seconds.
Permanent speed drop: 2.65 degrees after 1 minute 45 seconds.

Perfect combustion was maintained throughout these tests, no smoke being visible at the stack outlet.

The efficiency of 17.38 per cent could, of course, be further increased by the incorporation of an air preheater or of an exhaust steam turbine. The builders, however, deliberately refrained from the employment of such means, in order to retain the extraordinary simplicity of the plant which, combined with the relatively high efficiency, will in many practical cases be the determining factor in favor of the installation of a gas turbine. In addition to the simple and reliable design of the components of the unit: compressor, combustion chamber, turbine, there is the great advantage that *no cooling water is required*, making the location of the plant independent of any water supply. The low capital cost is also of importance, especially in regard to simplification of the foundations, small space requirements, and hence reduction in constructional costs.

Specially noteworthy are *the possibilities of raising the efficiency.*

(1) Heating the air, after it has been delivered by the compressor, by means of the exhaust gases. Here the gain realized depends on the area of the preheating surface.

(2) *Fractionalized combustion* consisting of a subdivision of the gas turbine into two or more stages, and a reheating of the gases in an intermediate combustion chamber by means of additional injected fuel, coupled with intercooling of the air compressor.

(3) Improvement of the efficiency at fractional load by *dividing the turbine into two units*, one driving the compressor, which may therefore run at the most favorable speed.

(4) Especially *raising the combustion temperature*, which will follow the improvement in heat resisting qualities of constructional materials, which may be expected to take place in the near future.

When it is recalled that for every improvement of 1 per cent in the efficiency of the turbine or of the compressor, due to improvements in the blade shape, the overall efficiency is raised by 4 per cent it will be appreciated that the gas turbine is full of promising possibilities meriting the immediate attention of prime mover designers and of industrial leaders.

MISCELLANEA OF INTEREST TO NAVAL ENGINEERS.

THE SPEED AND POWER OF DESTROYERS.—Reprinted from Shipbuilding and Shipping Record, February 1, 1940 issue, London.

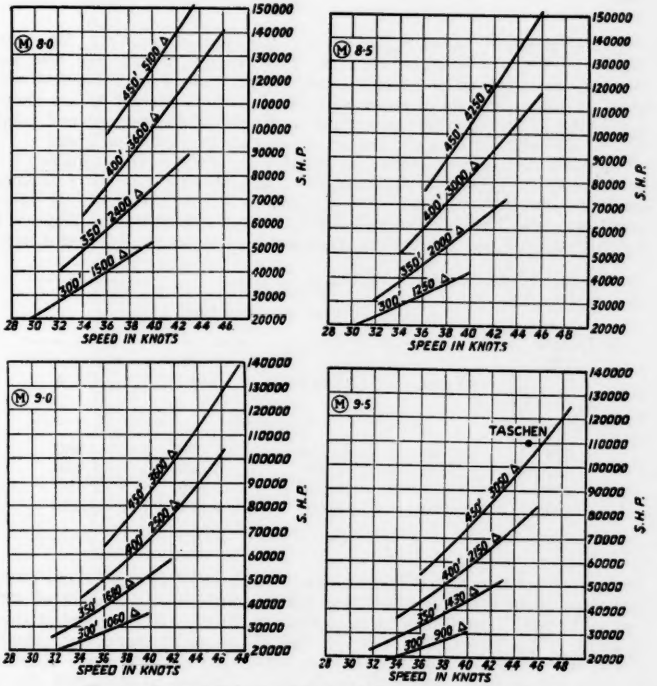
The speed record of 45 knots set up by the Italian-built Russian light cruiser *Taschen* is due to a given association of length, displacement and power installed. In British practice no endeavor has been made to increase the size of destroyers, so that they approach the dimensions of the *Taschen*, nor has the protection and armament given to light cruisers been reduced so that the small displacement of the *Taschen* can be attained and the great horsepower installed.

The length of this vessel is 453 feet, the breadth 45 feet, the displacement 3000 tons, the power of the propelling machinery 110,000 SHP., and the speed attained 45 knots. It is clear that this vessel is distinctly of the destroyer type, and although the speed is very high the performance is such as might have been expected from considerations of the results obtained with destroyers of more moderate size.

From an investigation of the results of basin and actual trials it is found that the determining factor in the performances of destroyers is the ratio of length and displacement. Form does not enter greatly into the standard of performance, although the length to breadth ratio may have some effect, nevertheless very minor in comparison with the length-displacement ratio effect. In general practice destroyers are given about 10 beams to the length and this ratio is fairly general in application. An attempt has been made to give a table of Admiralty constants taking into account the Ⓜ value (displacement-length ratio) and the speed length ratio $\frac{V}{\sqrt{L}}$. These values are as follows:

From the figures shown it is possible to arrive at very close approximations to the power necessary for any given vessel of the destroyer type knowing the waterline length and the displacement on trial.

Since most modern destroyers range from about 300 feet in length to a maximum of 450 feet, the length corrections may be neglected, since the Admiralty constants given above are assumed to be proper to vessels of 400 feet in length.



Diagrams showing S.H.P. required for given lengths, speeds and displacements

Ⓜ	Speed length ratios $\frac{V}{\sqrt{L}}$						
	1·7	1·8	1·9	2·0	2·1	2·2	2·3
8·0	145	147	148	150	152	155	161
8·2	152	152	153	155	158	161	166
8·4	157	157	159	161	164	166	170
8·6	162	163	164	166	168	172	176
8·8	167	168	169	172	174	176	180
9·0	171	172	173	175	178	180	183
9·2	175	176	177	179	181	184	188
9·4	178	179	180	182	185	188	191
9·6	181	182	183	186	188	191	194
9·8	183	184	186	189	191	194	197
10·0	186	187	189	191	193	197	199

$$Ⓜ = \frac{\text{Length}}{\text{Displacement}^{\frac{1}{3}}} 0{\cdot}3057$$

In the diagrams indicating the shaft horsepowers required for given lengths, speeds and displacements, the basic lengths chosen are 300 feet, 350 feet, 400 feet and 450 feet. These lengths have been associated with given Ⓜ values and the corresponding displacements arrived at are shown in the table at the top of the next column. From the diagrams several points may be noticed; for example, the great influence of weight on speed.

Suppose a destroyer of 350 feet in length with 50,000 SHP. available; if it were possible to get the displacement down to 1430 tons, a speed of 42 knots should be obtained, whereas with a displacement of 2400 tons, the speed would be reduced to 34½ knots. From the values given in the diagram it is possible to find the difference in speed which every 10 tons of weight incur.

Length in ft.	Displacements in tons			
	Ⓜ 8·0	Ⓜ 8·5	) 9·0	◯ 9·5
300	1,500	1,250	1,060	900
350	2,400	2,000	1,680	1,430
400	3,600	3,000	2,500	2,150
450	5,100	4,250	3,600	3,050

Again, consider the case of light cruisers. Many older British vessels of the type were given a length of 450 feet and a standard displacement of 4200 tons. The SHP. installed in these vessels was 40,000, and the speed attained 29 knots. This association of length and displacement corresponds almost exactly with that shown for the 450-foot vessel on the 8.5 Ⓜ value diagram. For a vessel of this length and displacement, 100,000 SHP. would give a speed of 39 knots, and if the displacement were reduced and the power increased to the 110,000 SHP., the *Taschen's* performance could be equalled.

FOR A VESSEL 350 FT. IN LENGTH WITH 50,000 S.H.P.

Displacement in tons	Speed in knots	Difference in displacement in tons	Difference in speed in knots	Difference in speed per 10 tons
1,430	42·1	—	—	—
1,680	39·6	250	2·5	0·10 knot
2,000	37·0	320	2·6	0·08 ,,
2,400	34·5	400	2·5	0·06 ,,

This question of weight and power is the determining factor in the speed problem and the efforts which are continuously being made to build lighter structures and secure greater power per unit of weight from the machinery installed are indicative of the importance of the weight problem.

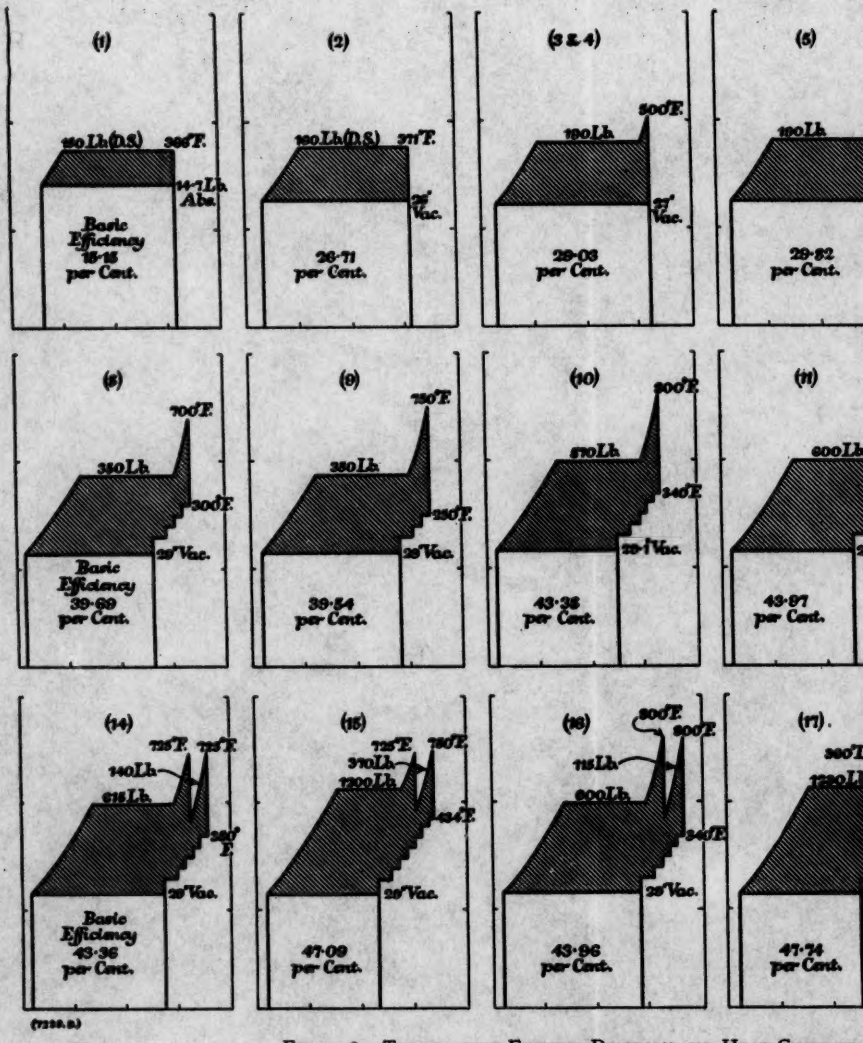


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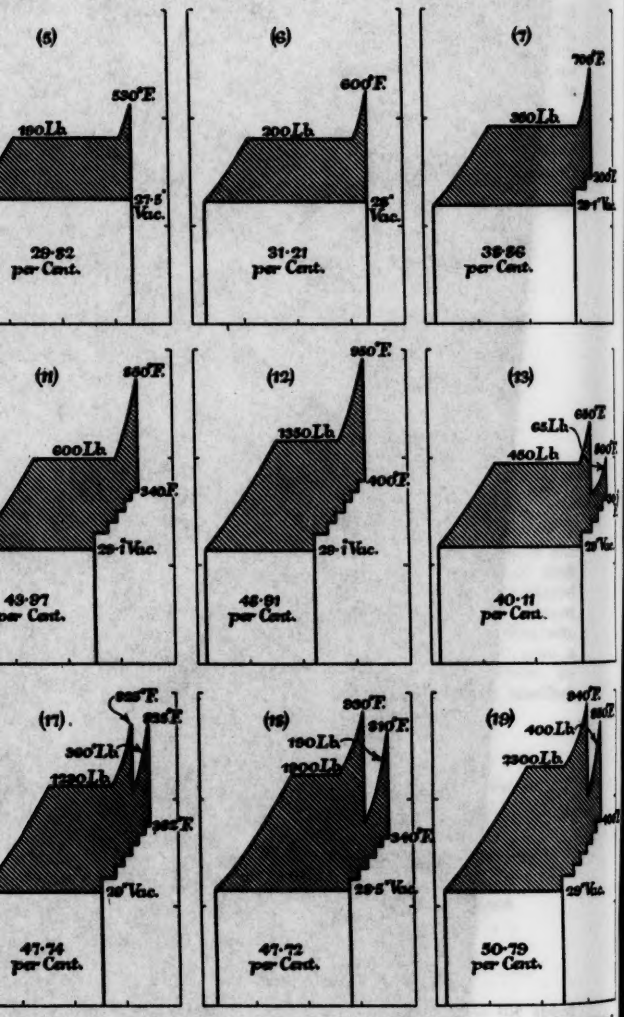
FIGURE 3.—TEMPERATURE-ENTROPY DIAGRAMS FOR HEAT CYCLE OF



T Cycle of Stations of Tables I and II.

FORTY YEARS ADVANCE IN EFFICIENCIES.—Excerpt from Engineering, London, April 22, 1940.

Engineering reports in abridged form the second part of the 26th Thomas Hawksley lecture delivered by Sir Thomas Pearce before the Institution of Mechanical Engineers in London on January 19, 1940. Of particular interest were the tables and diagrams, here reproduced, showing the rise in efficiencies of power stations in England throughout the past forty years. Figure 1 refers to power stations employing straight regenerative cycles and Figure 2 refers to stations employing a reheat cycle. Figure 3 shows the temperature-entropy for stations listed in Table I and Table II. An improvement in basic efficiency, over a period of forty years, from about 15 per cent to over 50 per cent is shown. Diagrams and tables only are reproduced here.

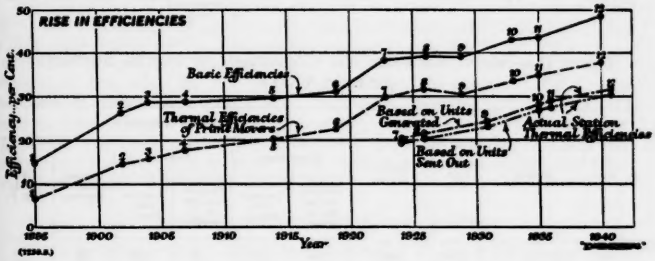


FIGURE 1.

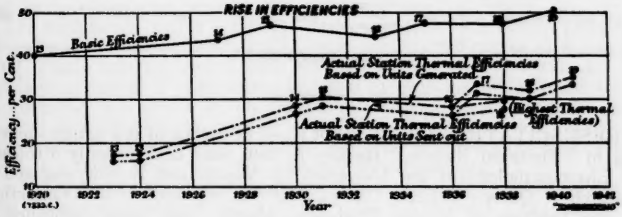

RISE IN EFFICIENCIES
Basic Efficiencies
Actual Station Thermal Efficiencies Based on Units Generated
(Highest Thermal Efficiencies)
Actual Station Thermal Efficiencies Based on Units Sent out
Efficiency—per Cent.
Year

FIGURE 2.

TABLE I.—SIZES AND TERMINAL CONDITIONS OF PRIME MOVERS, 1895-1940: STRAIGHT REGENERATIVE CYCLE.

No.	Date.	Station.	Size of Unit.	Maker.	Speed. R.p.m.	Terminal Conditions.			
						Pressure. Lb. per sq. in. gauge.	Temperature. Deg. F.	Vacuum. Inches of Mercury.	Feed Temperature. Deg. F.
1	1895	Manchester-sq...	350-kW turbine	C. A. Parsons	3,000	150	366 (dry sat.)	Atmos.	
2	1902	Stuart-street ..	1,500-kW (2,500 i.h.p. engine)	Yates and Thom	94	160	371 (dry sat.)	26	
3	1904	,, ..	5,000-kW (7,000 i.h.p. engine)	Wallsend Slipway	75	190	500	27	
4	1907	,, ..	6,000-kW turbine	Willans Siemens	1,000	190	500	27	
5	1914	,,	15,000 ,,	Howden Siemens	1,000	190	530	27·5	
6	1919	,,	20,000 ,,	Richardsons, Westgarth	1,500	200	600	28	
7	1923	Barton	27,500 ,,	Metropolitan-Vickers ..	1,500	350	700	29·1	200
8	1926	,, ...	41,000 ,,	,, ,, ..	1,500	350	700	29	300
9	1929	Deptford West.	35,000 ,,	,, ,, ..	1,500	350	750	29	250
10	1933	Battersea "A"	69,000 ,,	Met. Vickers-B.T.H. ..	1,500	570	800	29·1	340
11	1935	,, ,,	105,000 ,,	Metropolitan-Vickers ..	1,500	600	850	29·1	340
12	1940	,, "B"	100,000 ,,	,, ,,	Primary 3,000 Secondary 1,500	1,350	950	29·1	400

TABLE II.—SIZES AND TERMINAL CONDITIONS OF POWER STATIONS: REHEAT CYCLE.

No.	Date.	Station.	Size of Unit. kW.	Speed. R.p.m.	Terminal Conditions.					
					Initial Steam.		Reheat.		Vacuum. Inches of Mercury.	Final Feed. Deg. F.
					Pressure. Lb. per sq. in. gauge.	Temperature. Deg. F.	Pressure. Lb. per sq. in. gauge.	Temperature. Deg. F.		
13	1920	North Tees ..	20,000	2,400*	450	650	65	500	29	300
14	1927	Powerton	55,000	1,800	615	725	140	725	29	350
15	1929	Deepwater ..	53,000	Primary 3,600 Secondary 1,800	1,200	725	370	750	29	434
16	1933	Dunston "B" ..	50,000	1,500	600	800	115	800	29	340
17	1935	Port Washington..	80,000	1,800	1,230	825	360	825	29	382
18	1938	Brimsdown "A"	52,700	3,000	1,900	930	190	810	28·5	340
19	1940	Twin Branch ..	67,500	Primary 3,600 Secondary 1,800	2,300	940	400	850	29	400

* Later changed to 3,000 r.p.m.

RESUSCITATION FROM SHOCK.—The author of this article, appearing in "Electrical Review," January 5, 1940, was until recently Professor of Electropathology at the University of Vienna and is now engaged in research at Queen's College, Oxford. Extract reprinted from Transactions of the Institute of Marine Engineers, March, 1940, issue, published in England. Some new facts relative to reactions from electric shock are presented.

First-aid to a person who is apparently dead from electric shock is made more difficult by the preconceived idea among laymen that electricity acts like a poison or that a certain voltage or amperage will kill a man, so that from the first he considers the case as hopeless.

It is therefore of the utmost importance when giving instruction in first-aid to emphasize that the danger of an electric shock consists not only in the technical factors (*e.g.*, voltage, amperage, kind of current, earth circuit),

but also in the bodily and mental condition of the victim at the moment he receives the shock. However alarming the effect of an electric shock may be, recovery must never be regarded as hopeless, as experience gained from thousands of cases with a wide range of amperage has proved.

The question most often asked by doctors and engineers is: What voltage or amperage is fatal? The answer is a paradox: Any amperage or any voltage may, or may not, be fatal. For example, a man in Vienna was killed by d.c. at the very low voltage of 60, and another in Budapest by only 38 volts, whereas in Innsbruck a man was revived who received a shock at 125,000 volts.

The personal factor plays a great part. This is demonstrated every day by electricians who in the course of their work intentionally touch conductors charged at voltages that have frequently proved fatal in other cases. There are two reasons to account for this: (1) The electrician is healthy and is equipped, as the generality of people are, with a great tolerance for electricity. (2) His attention is fixed and he is prepared to receive the shock.

The evidence of the foregoing statements is that the majority of victims who have been revived evince no sign of internal injury to the brain or other vital organ, even when a great amperage has passed through the body. We have investigated the case of a victim whose hands had been totally destroyed but who did not faint even though he remained eight or nine minutes in circuit with a current at 2000 volts, showing that the vital organs had escaped injury. It is of paramount importance to the victim whether his attention is focused on the possibility of a shock or whether it takes him by surprise. Preparedness is therefore a protection against shock, the effect of which is usually decided by this factor.

ELECTRICAL CONTACT WHILE ASLEEP.

We were able to show that it was impossible to kill a deeply anaesthetized animal or cause it severe injury by means of a voltage that would have been fatal if it had been awake. This experiment has been confirmed by accidents occurring among laboratory workers. One man who fell asleep while watching a dynamo touched a bare wire. He was awakened by the contact and by a sensation of burning, but was not shocked or injured.

The explanation of this is that the action of electricity is chiefly functional. It causes a stimulus of such intensity as to paralyze the nerves supplying the vital organs bringing their actions out of gear, *i.e.*, to a deadlock, but the deadlock is reparable. Our assumption that the principal action of electricity is functional was proved by the experience we gained from accidents. Like a relay in mechanics, the function of the brain, heart, lungs and other vital organs did not always cease suddenly but after a varying interval of time or in a manner which was then unknown to pathologists.

Electric shocks may cause fatalities of four different types as follows:— (1) *Exitus momentaneus.*—The victim collapses suddenly. (2) *Exitus retardatus.*—He cries for help and after a few seconds or minutes falls unconscious. (3) *Exitus interruptus.*—The victim collapses suddenly, but after a few seconds gets up, speaks to his coworkers and then falls dead. (4) *Exitus dilatus.*—At first there appears to be no injury and the man may continue working. After an hour or more he may collapse and die suddenly.

An explanation of this may be found by post-mortem examination. In some cases paralysis of the brain was found to have been the cause of death. This indicates that the brain was the first organ to cease action.

In another case death was due to asphyxia or suffocation. In a third the primary cause of death was stoppage of the heart. In some persons there may be a predisposition to sudden death and a post-mortem examination may

reveal the sign of it. A less severe trauma even than an electric shock may be the cause of sudden death in such people.

In Austria we were able to convince employers of the necessity of excluding men who on examination were found to suffer from this predisposition from training as practical electrical mechanics. The great majority of post-mortem examinations, and we made five to six hundred, proved that the action of the heart, brain and vessels continued for minutes, in some cases for an hour, even though the doctor during that time was unable to detect any sign of life.

For instance, œdema of the brain or lungs in a man who was known previously to be healthy is definite proof that death was not instantaneous, but that there had been an interval to allow for the development of œdema, the man lying meantime between life and death in the state called suspended animation. A man rescued from drowning after not more than fifteen minutes is in the same state of suspended animation.

Why, if these findings are true, are so few victims of electric shock resuscitated? The reply is that first-aid as now practiced is often at fault, treatment not being begun early enough or carried on long enough. As in the resuscitation of a drowning man, time is the important factor. Every second counts. The first few seconds after an accident are decisive, but they are often wasted by transporting the victim, by washing his face or by leaving him alone while a doctor is summoned.

TOO VIGOROUS FIRST-AID.

Artificial respiration, as often carried out, is ineffective because the method is applied with too great force, resulting in the lowering or annihilation of the vacuum (negative pressure) in the thorax or even in physical injury. I was able to show by experiments on healthy rabbits, guinea-pigs, dogs and monkeys that artificial respiration applied with too great force might result in sudden death, due not to suffocation but to acute œdema of the brain and lungs.

Both methods of artificial respiration, that of Schäfer used in Great Britain and the United States and that of Sylvester used on the Continent, are founded on the compression of the thorax and have proved very effective.

I have proposed a new method of artificial respiration which is based not on compression but on an artificial dilation of the thorax, all direct pressure on the thorax being avoided. It is briefly explained by saying that an object such as a rolled-up coat is placed under the back between the shoulder blades, so as to raise the shoulders some inches from the ground. The shoulders are then seized and pushed forcibly backwards to the ground. The action is repeated every four or five seconds. By this maneuver the spinal column is stretched and the ribs expanded. This causes ventilation of the lungs and induces a suction action of the right side of the heart which improves the circulation.

The position of the patient permits the operator to note the first signs of returning life. A movement of the lips or larynx indicates the return of normal respiration and calls for the immediate cessation of artificial respiration. It is as harmful to continue artificial respiration after this takes place as to give it for too short a time.

This question has given rise to much discussion not only among laymen but also scientists; some say that artificial respiration must be carried on for one hour, others insist on two or three hours. I have always emphasized that the length of time is not a matter for subjective opinions; there are two objective signs which precisely determine the moment for ending artificial respiration: (1) The appearance of signs of returning life, the first being (*deglutition*), movement of lips. (2) Certainty of death having

occurred proved by the appearance of mortal spots (*livores mortis*). There is a good chance of recovery from shock if some form of first-aid is carried out promptly. The helper on the spot where the accident occurs is responsible for this immediate treatment until the doctor arrives. In some cases the latter may consider that the decisive factor in recovery of the patient is *venesection, lumbar puncture, benzin compresses,* and other treatment in addition to artificial resuscitation.

There should be a first-aid corps in every factory with practical training in artificial respiration, and knowledge of how to extricate the victim from the electric circuit.

During the last war electrically-charged wire or overhead wires liable to be hit by aircraft were the cause of electric shock or fire and many soldiers were killed. The Austrian Ministry of War supplied the troops specially engaged in electrotechnics with an "electrosaving-knapsack" constructed from my design. These men having had practical instruction in the use of the tools and contents of this knapsack were able to improvise means for saving the victims of electric shock.

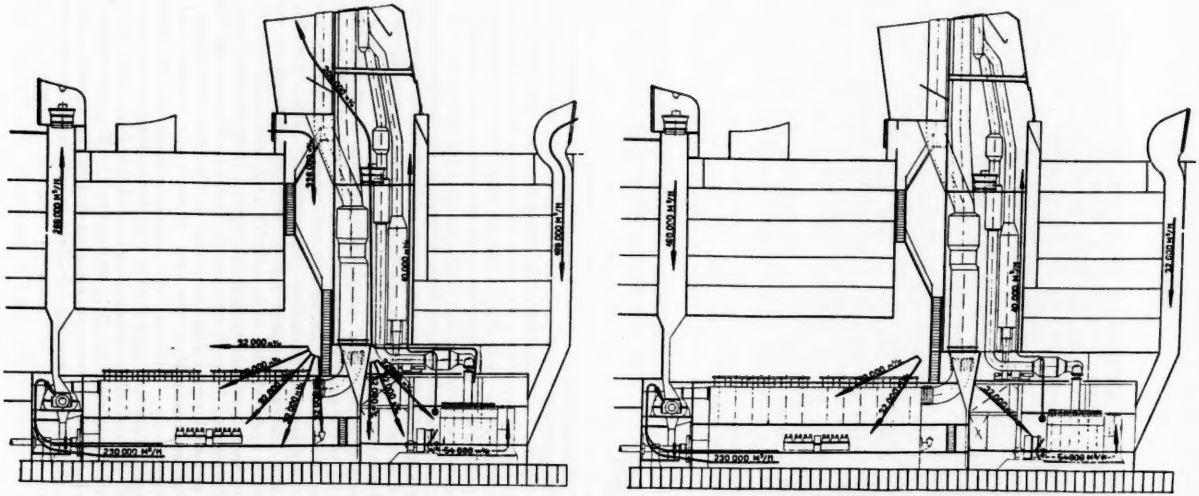
Resuscitation from an electric shock may perhaps not be classed among the arts, but it is a handicraft which requires knowledge of technique and practice.

ENGINE-ROOM VENTILATION IN LINERS.—Reprinted from the June, 1940, issue of The British Motorship, published by The Temple Press, London, England.

It must be admitted that in some of the earlier motor passenger liners the ventilation arrangements in the main and auxiliary engine-rooms were a little sketchy. Such a state of affairs did not persist for long, and in modern vessels of the class, particular care, with successful results, has been taken to meet the widely varying conditions of atmospheric temperature and humidity. Consequently, in most big passenger liners the conditions under which the engine-room staffs work could scarcely be improved.

The Netherland Steamship Co.'s liner *Oranje* is the highest-powered mercantile motor ship afloat, hence the design and arrangements of the ventilating systems in the machinery compartments have a particular interest. It may be recalled that there are two engine-rooms, and that in the main machinery space three 12,500 BHP. single-acting two-stroke Sulzer engines are installed, whilst in the auxiliary engine-room, and separated by a water-tight bulkhead, are five 1200-Kw. Diesel-engined generators, besides which are numerous motor-driven pumps in both compartments. Moreover, the adjoining refrigerating machinery space and a large compartment in which the distilling plant is installed have to be suitably ventilated. The *Oranje* in her service has to meet extreme conditions varying from tropical heat in the Dutch East Indies and the cold weather of Northern Europe, and these conditions have to be allowed for in the design of the ventilating plant.

The illustrations below show the flow of air under summer and winter conditions, respectively. Although the main propelling engine exhausts are very well lagged, there is a good deal of heat transmitted to the main engine-room through the cylinders, framing and crankchambers, since the average temperature of the lubricating oil is 50 degrees C. The scavenging blowers are in an enclosed compartment at the after-end of the engine-room and the suction from them has a very helpful effect in ventilation, as the amount of



Disposition of engine-room ventilation under tropical conditions (left) and under winter conditions (right).

air required is normally 230,000 cubic meters per hour for the engines. The air is drawn through six openings in the floor of the scavenging pump chamber and the main supply of air enters the engine-room through openings in the casing at the foot of the funnel on the top deck, this amount being estimated at 336,000 cubic meters per hour. In addition are 10 electrically driven ventilators which deliver air from the same source to various ducts throughout the engine-room. In these ducts are louvres so that the discharge can be regulated to any extent, whilst all of the motors (which are short-circuited, 220/127-volt a.c. machines) can be arranged to run at half speed, which gives sufficient variation of output.

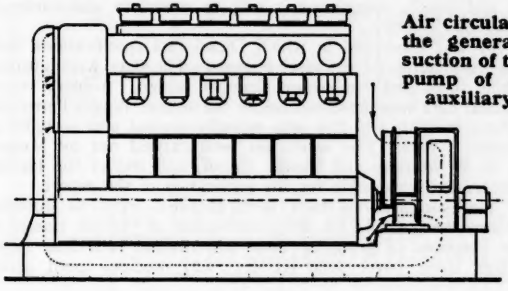
At the after end of the engine-room is a duct leading to an outlet on the top deck, and under tropical conditions the two ventilating fans at the top are both in operation drawing air from the scavenging blower compartment, and therefore from the engine-room, the normal suction under tropical conditions being 266,000 cubic meters per hour. With this arrangement, it is claimed (and we can confirm the claim from experience during trials) that there are no corners of the engine-room in which "dead" air can linger. Either one or both of the after exhaust ventilators can be stopped, to adjust the temperature and ventilation in the engine-room, but there is no regulation for the quantity of air admitted to the scavenging-air blower compartments.

Under winter conditions, naturally, a smaller quantity of air is required from the atmosphere, although approximately the same amount is, of course, needed for the scavenging-air blowers. In order to meet this condition, the ventilating fans at the top of the after vertical trunk are reversible, and in winter they deliver air to the scavenging air blower compartment, instead of abstracting it from it. Moreover, some of the ventilators delivering air to the ducts already mentioned are not required, and the total amount of air is made up as shown in the illustration. For heating the engine-room in port under winter conditions, steam equipment is provided. Under tropical conditions, in port the required number of the 10 ventilating fans are placed in operation, also one of the ventilators at the top of the aft vertical trunk.

AUXILIARY ENGINE-ROOM.

Ventilating conditions in the auxiliary engine-room under tropical and winter conditions, respectively, when at sea do not vary quite so much as in the main engine-room. There is always a relatively large amount of heat generated by the five 1200-Kw. Diesel-engined dynamos and the auxiliary motors, and the compartment is low and comparatively small.

The main delivery of air into the auxiliary engine-room to the extent of 180,000 cubic meters of air per hour in the tropics and 32,000 cubic meters



Air circulation around the generator, to the suction of the scavenge pump of one of the auxiliary engines.

per hour under winter conditions, takes place through a vertical shaft at the forward end of the compartment. The suction from the scavenge pump at each engine draws the air over the dynamo and under the bedplate, which is enclosed in the manner shown herewith. In the funnel shaft are two suction ventilators with a capacity of 200,000 cubic meters of air, the diameter of the ventilators being 1800 millimeters. In the tropics these draw the air from the top of the auxiliary engine-room and discharge through the funnel into the atmosphere. Additional ventilation is provided from four ventilating fans, one of which delivers air near the switchboard and another to the transformer compartment.

As mentioned earlier, there are two additional chambers that require ventilation close to the auxiliary machinery room, one of which houses the refrigerating machinery and the other the distilling plant. In the latter, there is piping in which circulates distilled water under a slight vacuum. In order to avoid the possibility of oil-contaminated air from the auxiliary engine-room entering this compartment and the piping, the room has to be kept under a pressure slightly above that of the atmosphere, and there are two ventilators at the bottom of the tunnel shaft which supply air for this purpose to the distilling machinery room.

In the refrigerating compartment, the conditions are reversed. The ammonia is under pressure and it is necessary that any gas which may leak should be abstracted. For this reason, therefore, an exhauster fan is provided.

A STUDY OF FUEL SPECIFICATIONS.—Reprinted from the June, 1940, issue of The British Motorship, published by the Temple Press, London, England.

All Diesel engine users, and some manufacturers, are of opinion that the future technical development of the engine must be along lines so that it becomes less discriminatory in the grade of fuel on which it can successfully operate. At present, the majority of manufacturers somewhat naturally prefer that the fuel oil employed should be of reasonably good characteristics, and in many cases they have specified the grade which they recommend.

Even if poorer grades of oil have to be utilized in the future in Diesel machinery, as seems probable, there will still remain standards, and it is, therefore, highly desirable to have the views of oil engine builders on the suggested limit in specifications for fuel, whilst the opinions of users are also of importance. Some investigatory work of this nature has been carried out in America and details concerning it will be useful to manufacturers and shipowners in this country.

In 1935 a survey was conducted in which Diesel fuel specifications were collected from 38 manufacturers of Diesel engines. The data were grouped according to engine speed and combustion chamber design. Another report based upon additional data was presented before the second World Petroleum Conference at Paris in 1937, and this was recently revised and brought up to date on October 1, 1939. The work has been carried out by Messrs. W. H. Hubner, G. B. Murphy and Gustav Egloff, and one of the leading American oil concerns is responsible for the publication of the report.

Recommended fuel specifications have been obtained from 60 different engine manufacturers, responsible for the construction of various classes of engines, and for purposes of grouping these are divided into four types, namely:—(1) High speed (over 1500 RPM.); (2) medium speed (from

Table I.—Manufacturers' Recommended Fuel Specifications for Low-speed Engines (Below 500 r.p.m.)

Engine Make	Viscosity S.U. @ 100° F. Seconds Min.	Viscosity S.U. @ 100° F. Seconds Max.	Gravity °A.P.I. Min.	Gravity °A.P.I. Max.	Sulfur % Max.	Hard Asphalt % Max.	Conradson Carbon Residue % Max.	Ash % Max.	Water and Sediment % Max.	Flash Pt. °F. Min.	Pour Pt. °F. Max.	Heat Value B.t.u./lb. Min.	Distillation Characteristics, °F. 10% Pt.	Distillation Characteristics, °F. 90% Pt.	Distillation Characteristics, °F. E.P.	Ignition Quality Min.
Direct Injection:																
1. Alco-locomotive	–	250	–	–	2.0	–	3.0	0.04	0.6	150	35(b)	–	–	–	–	30 Cetane-delay(3)
2. Alco-4 cycle	–	250	–	–	2.0	–	3.0	0.04	0.6	150	35(b)	–	–	–	–	30 Cetane-delay(3)
3. Alco-Sulzer	–	250	–	–	2.0	–	3.0	0.04	0.6	150	35(b)	–	–	–	–	30 Cetane-delay(3)
4. Alliance	–	–	24	38	–	–	–	–	–	–	–	18,600	–	–	–	–
5. Allis-Chalmers(m)	(No specific fuel specifications)															
6. Atlas Imperial	35	60	–	–	0.75	–	0.5	0.05	0.1	–	–	–	–	–	–	35 Cetene–C.C.R.
7. Atlas Polar	–	75	–	–	2.0	1.0	–	0.02	1.0	150	35	17,700	–	–	–	–
8. Buckeye	35	–	30	32	0.3	–	–	–	Trace	190	0	–	–	–	–	–
9. Busch-Sulzer	35	70	–	–	1.5	–	0.5	0.02	0.1	150	35(b)	–	–	–	–	35 Cetane-delay
10. Busch-Sulzer(m)	–	250	–	–	2.0	–	3.0	0.04	0.6	150	35(b)	–	–	–	–	30 Cetane-delay
11. Cooper Bessemer	50	100	–	–	2.0	–	2.0	0.05	0.5	150	–	–	–	–	–	40 Diesel Index
12. Deutz	(Any customary fuel specification)															
13. Fairbanks, Morse	35	75	–	–	1.0	–	2.0	–	1.0	150	15(b)	19,000	–	650 max.	–	45–60 cetane
14. General Motors (Cleveland)	35	70	–	–	1.0	–	0.15	0.02	0.1	150	35(b)	–	–	–	–	40 Cetane-delay
15. Hooven, Owens	75	85	24	–	2.0	–	1.5	–	1.0	150	12	19,000	–	–	–	–
16. Ingersoll-Rand	–	80	–	–	1.0	–	0.5	–	1.0	150	–	19,000	–	–	–	45 Cetene–C.C.R.
17. Nelseco	–	250	–	–	2.0	–	3.0	0.04	0.6	150	(b)	–	–	–	–	30 Cetane-delay
18. Nordberg(q)	30	80	–	–	1.5	0.5	–	0.05	0.5	150–200	0	19,000	–	–	–	–
Nordberg(m)	80	300	–	–	2.5	–	–	–	2.0	150–200	0	19,000	–	–	–	–
19. Page Engineering	42	–	26	36	1.0	–	0.2	Trace	1.0	150	(b)	18,500	(I.B.P. 400 max.)	50% @ 550 max.	700 max.	50 Cetane-delay
20. Reid(o)	(No specific fuel specifications)															
21. Ruston	(No specific fuel specifications)															
22. Standard	–	–	24	–	–	–	–	–	–	–	–	–	–	–	–	–
23. Sun-Doxford	45	150	14	19	0.6	–	10.0	0.15	1.0	150	0	18,600	–	–	–	–
24. Sun-Junker	App.	210	17	21	0.6	–	2.0	0.08	1.0	150	0	19,000	–	–	–	–
25. Superior	40	50	24	36	0.5	–	0.07%	Nil	Nil	150	–	19,000		Satisfactory	–	(40)
26. Union	(Commercial Diesel fuel)															
27. Washington Iron	–	–	24	35	–	–	–	–	–	–	–	–	–	–	–	–
28. Weber	–	–	28	–	–	–	–	–	–	–	–	19,000	–	–	–	–
29. Western	40	80	24	29	0.5	–	–	–	–	150	–	–	–	–	–	–
30. Worthington	30	50	28	32	1.5	–	1.0	0.01	0.1	150	0	18,500	–	–	–	45 Cetane-delay
Pre-combustion:																
31. Anderson	App.	37	28	36	0.5	–	0.02	Nil	Nil	150	–	–	app. 450	app. 635	app. 690	140–150° F. Aniline Pt.
32. Bolinders	–	–	28	36	0.5	–	–	Nil	–	–	–	–	–	–	–	–
33. Chicago Pneumatic	32	70	–	–	2.0	–	0.5	0.02	0.2	150	(b)	–	–	–	–	35 Cetane-delay
34. Wolverine	–	–	24	32	0.5	5.0(n)	1.0	0.05	0.3	125	–	18,500	–	–	–	–
Separate-chamber																
35. De La Vergne	40	80	25	–	1.0	–	1.5	0.02	0.5	150	35(b)	–	(I.B.P. 375 min.)	700 max.	–	40 Cetane-delay
36. Venn-Severin	40	90	26	35	1.5	–	2.0	0.05	1.5	150	15	18,500	–	–	700 max.	40–60 Cetane
Air-cell:																
37. M.A.N.	App.	55	25	–	0.2	0.7	1.8	–	1.0	150	0	19,150		60% @ 662 max.		(4)

1000-1500 RPM.); (3) medium speed (from 500-1000 RPM.); and (4) low speed (up to 500 RPM.).

The classification is further arranged in terms of combustion chamber design (direct injection, pre-combustion chamber, separate chamber, air cell and energy cell).

For marine work the engines that mainly come into question are those operating below 500 RPM., and Table I gives the manufacturers' recommended fuel specifications for the engines of this class, which were brought under consideration. In Table II the recommended specification for each class of engine is represented by the average of a number of manufacturers in each class; in the same table the recommended fuel specification for the particular class of engine as outlined by the American Society of Testing Materials is also given.

There is surprisingly little difference shown in the specifications outlined by the manufacturers of low-speed engines and those of the high-speed type, except, of course, in the matter of viscosity. As might be anticipated, gravity appears to be of little importance in the specification.

There are so many methods of determining ignition quality, especially in the United States, that comparisons are somewhat difficult to make. But so far as they are possible, the demands of the manufacturers of various classes of engine do not appear to vary widely.

In the report in which the various tables are presented, and, incidentally, the names of all of the manufacturers are given, some of the statements of the builders are of particular interest. The variety of views is a little surprising, as the following extracts indicate:—

"The fuel must be wholly distilled." (Allis-Chalmers Manufacturing Co.)

"We have used almost all grades from kerosene to lubricating oil without loss in efficiency or power output." (Continental Motor Co.)

"Our engine takes what it can get." (Bolinders Co., Inc.)

"Fuel oil should be a neutral distillate petroleum oil and not a mixture of light oil and heavy residue." (Northill Co., Inc.)

"The fuel can be a cracked residual, a blend or a straight run distillate." (Buda Co.)

"Fuel oil should consist of petroleum hydrocarbon distillate containing no cracked distillate nor residuum."

"Diesel fuels should be a petroleum distillate."

IGNITION QUALITY.

On the subject of ignition quality there is a similar difference of opinion.

"We do not believe in the importance of cetane numbers for a good and thoroughly constructed engine. We fear that the cetane number will be only an excuse for engine builders whose Diesel engines do not run with normally refined oils." (Bromfield Manufacturing Co.)

"Some fuels cannot be burned at all in high-speed engines because of their low ignition quality." (Venn-Severin Machine Co.)

"The ignitibility of fuel oil cannot be determined by the usual characteristics of physical state of oil, such as gravity, viscosity or color, except that oils of low viscosity and low gravity are indicative of a hard fuel oil to burn." (Hercules Motor Corporation.)

"There is unquestionably a difference in the ability of various types of Diesel engines to handle fuels of low cetane value. For the present, we feel that it is desirable to keep above 45 cetane; as our knowledge and experience increase, we may be able to lower this. Our cetane number determination is made by the critical compression ratio (C. C. R.) method, and

Table II.—Manufacturers' Recommended Fuel Specifications and A.-S.T.M. Fuel Oil Classification for High, Medium High, Medium and Low-speed Engines.

	High Speed (above 1500 r.p.m.) 14 Manufacturers		ASTM	Medium High-Speed (1000–1500 r.p.m.) 31 Manufacturers		ASTM	Medium Speed (500–1000 r.p.m.) 36 Manufacturers		ASTM	Low Speed (below 500 r.p.m.) 37 Manufacturers		ASTM
	Spread	Average	No. 1-D	Spread	Average	No. 1-D	Spread	Average	No. 3-D	Spread	Average	No. 4-D
1. Viscosity S.U. at 100°F.:												
a) Seconds, min.	30	34	33	30	34	33	30	37	33	30	42	–
b) Seconds, max.	100	51	50	100	60	50	100	64	70	300	125	250
2. Gravity, °A.P.I.:												
a) Minimum	26	28	–	22	27	–	22	27	–	14	25	–
b) Maximum	40	36	–	40	35	–	40	36	–	38	32	–
3. Sulfur, % max.	0.5–1.5	8.1	1.5	0.1–2.0	0.9	1.5	0.2–2.0	0.96	1.5	0.2–2.5	1.25	2.0
4. Hard Asphalt, % max.	0.01–1.0	0.45	–	0.05–1.0	0.47	–	0.3–1.0	0.63	–	0.5–5.0	1.5	–
5. Conradson Carbon, % max.	0.03–2.0	6.6	0.2	0.03–2.0	0.55	0.2	0.03–2.0	0.65	0.5	0.02–10.0	1.8	3.0
6. Ash, % max.	0.01–0.1	0.03	0.02	0.01–0.1	0.03	0.02	0.00–0.1	0.03	0.02	0.00–0.05	0.04	0.04
7. Water & Sediment, % max.	0.00–1.0	0.26	0.05	0.00–1.0	0.3	0.05	0.01–1.0	0.32	0.1	0.01–2.0	0.6	0.6
8. Flash Pt., °F. min.	120–175	150	Legal	120–150	148	Legal	149–190	154	150	125–200	150	150
9. Pour Pt., °F. max.	–	(a)	35(b)	–	(a)	35(b)	–	(a)	35(b)	–	(a)	35(b)
10. Heat Value, B.t.u./lb., min.	18,500–19,300	18,900	–	17,800–19,500	18,900	–	17,850–19,300	18,865	–	17,700–19,150	18,716	–
11. Distillation Characteristics, °F.:												
a) I.B.P., max.	–	–	–	–	–	–	–	–	–	–	–	–
b) 10% pt., max.	425–475	457	–	425–475	458	–	450–510	470	–	–	450	–
c) 90% pt., max.	580–675	633	–	580–690	643	–	580–690	663	–	635–700	662	–
d) E.P., max.	650–725	700	–	600–725	686	–	600–725	693	–	690–700	697	–
12. Ignition Quality*:												
a) Cetene No., C.C.R., min.	–	–	–	–	45	–	40–45	43	–	35–45	40	–
b) Cetene No., delay, min.	–	–	–	–	42	–	–	–	–	–	–	–
c) Cetane No., C.C.R., min.	–	–	–	–	40	–	–	–	–	–	–	–
d) Cetane No., delay, min.	38–50	47	45	35–60	47	45	35–60	43	35	30–60	38	30
e) Diesel Index No., min.	–	–	45	–	–	45	–	–	30	–	40	20
f) Viscosity-Gravity No., max.	–	–	0.86	–	–	0.86	–	–	0.89	–	–	0.91
g) Boiling Point-Gravity No., max.	–	–	188	–	–	188	–	–	195	–	–	200
h) S.I.T., min.	–	low	low	–	–	–	–	–	–	–	–	–
i) Aniline Point, °F., min.	–	–	–	–	–	–	–	–	–	140–150	145	–

NOTES.

does not correlate exactly with ratings made by the delayed period method." (Waukesha Motor Co.)

"Use of a fuel of low ignition quality results in difficult starting, rough running, high maximum combustion pressures and probably excessive combustion temperatures. These conditions are all very harmful in an engine and can result in distorted and stuck injector spray nozzles, pitting of exhaust valve seats, rapidly sludged lubricating oil resulting in worn piston rings, grooves and cylinder bores." (Peet and Powers, Inc.-Lister and Ruston.)

VISCOSITY.

There are various interesting comments on the question of viscosity.

"Viscosity must be controlled within reasonable limits if we are to have any control over the time of injection and the quantity of fuel injected.

"Highly refined oil is often dry and should be enriched with a heavy oil having some lubricating value." (Page Engineering Co.)

"We have demonstrated that 200 viscosity can be taken care of, but it is our oplnion that the trouble experienced when the viscosity is in that range does not pay for the slight advantage in cost of the cheaper fuel, and for this reason we have preferred to hold it down to a maximum of 100." (The Cooper Bessemer Corporation.)

"We have had some trouble with the fuel oil on account of, it is claimed, lack of lubricity.

"Viscosity has to be within a certain range on account of opposing factors. Against an oil of high viscosity, one may cite that foreign matter has less chance to settle out and is more likely to be carried into the fuel injector. Against oil of very low viscosity is the liability of excessive leaking past the sealing portion of the fuel pump plunger and spray nozzle. Oils of very low viscosity are generally deficient in lubricating property, and if they are used it becomes necessary to add a proportion of lubricant to them to insure satisfactory lubrication of the pump plungers and sprayer valves." (Peet and Powers, Inc.)

CONRADSON CARBON.

Again, in regard to the question of Conradson carbon, we find a wide diversity of views.

"We consider the Conradson carbon content as probably the most important item in a fuel specification." (Associated Equipment Co.)

"We have not been able to trace any of our particular difficulty to the carbon, as indicated by the Conradson carbon test." (Chicago Pneumatic Tool Co.)

"Conradson carbon we know is detrimental. We know our engines will not operate satisfactorily with around 5 per cent Conradson carbon and we know that we have engines operating satisfactorily with 2 per cent." (Continental Motors Corporation.)

"Conradson carbon residue gives no reliable indication of carbon formation because of the difference in combustion conditions, i.e., a deficiency of oxygen in the laboratory test and air excess in the engine test.

"Using a modified test, a residue above 1 per cent has been found troublesome in practice." (Waukesha Motor Co.)

"Using a modified test a residue up to 10 per cent is satisfactory, the residue being that portion of oil remaining in the cup after being evaporated to a temperature of 527 degrees F. for 120 hours." (General Motors Corporation.)

On the subject of flash point we find two directly contradictory views, one of which is that "flash point comes first in importance" and the other "No Diesel engineers believe this factor is important."

These comments were extracted from a large number published in the "report" and are of great interest, even though they are, in most cases, from manufacturers who are chiefly concerned with the building of high-speed Diesel engines. They indicate a rather remarkable variety of views, but as the authors point out in their summary, manufacturers must, in the end, design their engines to operate satisfactorily on fuels of average ignition quality.

BOOK REVIEW.

MOBILIZING CIVILIAN AMERICA, BY HAROLD J. TOBIN AND PERCY W. BIDWELL, PUBLISHED BY COUNCIL ON FOREIGN RELATIONS, 45 EAST 65TH ST., NEW YORK, PRICE $2.75.

This volume, published on May 27, 1940, presents a great deal of material for serious thought. It is of such timely interest that this review may not appear in time to announce the book before the application of its thoughts become necessary.

The book is centered about the War Department's Plan for Industrial Mobilization. However, its fairly thorough recountal of World War experience and its discussion of the Plan leave the reader with the impression that, at best, a series of problems have been presented and the answers are yet to be found.

A thought provoking volume well worth study by those who are to be military or civilian leaders during a future war in which the United States may become involved.

ASSOCIATION NOTES.

Change in Office of Secretary-Treasurer: Because of detachment from duty in the City of Washington, Lieutenant Commander Guy Chadwick, U. S. Navy, tendered his resignation as Secretary-Treasurer of the Society at a meeting of the Council held on April 10, 1940, effective on April 22, 1940. The resignation was accepted with much regret and a unanimous expression of thanks and appreciation to Lieutenant Commander Chadwick for his efficient work.

Lieutenant Commander James E. Hamilton, U. S. Navy, was unanimously appointed Secretary-Treasurer to fill the unexpired term of Lieutenant Commander Chadwick.

Naval Messages of Interest to the Society: A number of Naval messages of interest or importance to the Society have been broadcast to the Service since the last issue of the JOURNAL went to press. Little comment is necessary. Each is self explanatory.

On May 24, 1940:

> "The President has approved the report of the Construction Corps Selection Board. Officer recommended for advance to Rear Admiral is Captain Henry Williams."

The Society's President will be advanced to the rank of Rear Admiral on October 1, 1940.

On June 24, 1940:

> "Legislation completed establishing Bureau of Ships combining all duties of former Bureau of Engineering and Construction and Repair. Rear Admiral S. M. Robinson, U. S. N., appointed Chief of new Bureau. Rear Admiral A. H. Van Keuren (CC) U. S. N., Assistant Chief. Captain Henry Williams (CC) U. S. N., detailed as Administrative Assistant."

On June 28, 1940:

"Under date of June 25, 1940, law enacted which abolishes Construction Corps of Navy and transfers officers on active list of Construction Corps on date of Act to line of Navy, designated for Engineering Duty Only."

MEMBERSHIP.

The following members have joined the Society since the publication of the May, 1940, JOURNAL:

NAVAL.

Adams, Harry A., Jr., Lieutenant, U. S. N. R., Bay State Apartments, 1701 Massachusetts Ave., N. W., Washington, D. C.

Casey, Robert F., Lieutenant, U. S. C. G., 68 Johnson Ave., Bloomfield, N. J.

Cockaday, L. M., Lieutenant Commander, U. S. N. R., Commanding Section 3, Naval Communications Reserve, Third Naval District. Residence, 547 Second Ave., North Pelham, N. Y.

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Den Hartog, J. P., Lieutenant Commander, U. S. N. R., Pierce Hall, Harvard University, Cambridge, Mass.

DuMont, Paul J., Lieutenant, U. S. N. R., Port Engineer, Grace Lines, New York, N. Y. Residence, 20 Hornblower Ave., Belleville, N. J.

Fairman, Richard H., Lieutenant, U. S. C. G., 1353-C McCutcheon Road, Richmond Heights, Mo.

Katz, Morris Carl, Lieutenant Commander, U. S. N. R., U. S. Sales Representative, Eastern Division, Eagle Electric Mfg. Co., Brooklyn, N. Y. Residence, 5479 Montgomery Ave., Philadelphia, Pa.

Morris, Robert L., Lieutenant, U. S. Navy.

Seielstad, Harold D., Lieutenant, U. S. C. G., 1017 South Berry Road, Webster Groves, Mo.

Simpson, R. T., Lieutenant, U. S. N. R., Turbine Division, General Electric Company, Schenectady, N. Y.

Williams, Edward Ogden, Ensign, U. S. N. R., S. S. *Gargoyle,* Socony-Vacuum Oil Co., Inc., 26 Broadway, New York, N. Y.

Williams, J. H., Ensign, U. S. Coast Guard.

TRANSFERRED CIVIL TO NAVAL.

Fowler, Thomas P., Lieutenant, U. S. C. G., Coast Guard Headquarters, Washington, D. C.

CIVIL.

Barron, George E., Chief Engineer, Maryland Dry Dock Co., Baltimore, Md. Residence, East Greenwood Road, Linthicum Heights, Md.

Gale, Albert G., General Electric Company, Lynn, Mass.

Glassie, Don Caffery, Head, Don Caffery Glassie Co., Munsey Building, Washington, D. C.

Gleitz, Herbert, President, Marquette Metal Products Company, Cleveland, Ohio. Residence, 1145 Galewood Drive, Cleveland, Ohio.

Guthrie, F. P., District Communications Manager, RCA Communications, 1112 Connecticut Ave., N. W., Washington, D. C.

Harrell, Fred E., Assistant Chief Engineer, Reliance Electric and Engineering Co., 1088 Ivanhoe Road, Cleveland, Ohio.

Hohl, Henry D., Vice President of Manufacturing, Triangle Conduit and Cable Co., New York, N. Y. Residence, 25 Kenmore Road, Valley Stream, L. I., N. Y.

Main, Archibald M., Vice President and General Manager, Bath Iron Works Corporation, Bath, Me. Residence, 30 Garden St., Bath, Me.

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The following Erratum was received too late for insertion in the usual place:

ERRATA

Issue of November, 1939.

Article " The Collapse of Hard and Soft Copper Tubes Under External Pressure," by Maurice G. Steele.

On page 535, in Figure 18, the curve on the right which is marked " 2½" O. D." should read " 3" O. D."